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PATENTS
2002-1026

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE
BEFORE THE BOARD OF PATENT APPEALS AND INTERFERENCES

In re application of

VAN DER SLUIS et al. Conf. 9054

Appl. No. 10/661,541 Group 3682

Filed September 15, 2003 Examiner V. Johnson

DRIVE BELT AND CONTINUOUSLY VARIABLE TRANSMISSION WHEREIN SUCH IS UTILISED

APPEAL BRIEF

MAY IT PLEASE YOUR HONORS: November 2, 2007

(i) Real Party in Interest

The real party in interest in this appeal is the Assignee, Van Doorne's Transmissie B.V. of Tilburg, The Netherlands.

(ii) Related Appeals and Interferences

Neither the appellants, appellants' legal representative nor the assignee know of any other prior or pending appeals, interferences or judicial proceedings which may be related to, directly affect or be directly affected by or have a bearing on the Board's decision in the pending appeal.

(iii) Status of the Claims

Claims 1-11 are pending and rejected. The final rejection of claims 1-11 is being appealed.

iv) **Status of Amendments**

There have been no amendments to the claims since the Final Official Action mailed May 18, 2006 (the "Official Action").

(v) **Summary of Claimed Subject Matter**

As per independent claims 1, 4, and 7, the invention is a drive belt (3) for a continuously variable transmission ("CVT") that comprises a continuous band (11), having a radially inwardly oriented surface (12) and a radially outwardly oriented surface (13), and an array of transverse elements (20) engaging the continuous band (11) such that the elements (20) may slide along a longitudinal direction thereof (specification page 5, lines 5-22; page 5, line 35 to page 6, line 10).

Claims 1, 4, and 7 each further require the continuous band (11) be curved in a transverse direction at a non-infinite crowning radius of curvature Rcrown and provided with an internal residual stress distribution defining a curling radius of curvature Rcurl at which a continuous band (11) would be curved in its longitudinal direction when cut. See Figure 6 and specification page 4, lines 22-24; page 8, lines 12-14.

Claim 1 recites the pre-bending factor equation. Specification page 8, lines 32-35 discloses that, the pre-

bending factor $f_{PB} = (\delta i + \delta o)/\delta o$ (7), wherein, δi is the largest perpendicular distance in the radial direction between a neutral line NL in the cross section of the continuous band (11) where the stress due to pure longitudinal bending would be zero and the radially inner most surface (12) of the band (11), and δo is the largest perpendicular distance in the radial direction between the said neutral line NL and the radially outer most surface (13) of the band (11).

The pre-bending factor recited in independent claim 7 is $f_{PB} = \{(f_i/f_o)\cdot\delta i + \delta o\}/\delta o$, wherein, f_i is a stress factor defining the relative increase of the maximum tension stress at the radially inner most surface (12) due to anticlastic bending when the band (11) is bent straight, and f_o is a stress factor defining the relative increase of the maximum tension stress at the radially outermost surface (13) due to anticlastic bending when the band (11) is longitudinally curved at the said minimum radius of curvature Rmin. This maps to specification page 9, beginning at line 11 concerning anticlastic bending that is discussed in, *inter alia*, the European patent applications EP-A-1.111.271 and EP-A-0.905.830. The stress factors f_i and f_o map to specification page 9, beginning with line 34 disclosing comparing the maximum tension stresses in the continuous band 11 when it is straightened and when it is bent at the minimum radius of

curvature Rmin that are obtained with Finite Element Method (FEM) calculations wherein the anticlastic bending phenomenon is included with the results obtained with equation (6).

The pre-bending factor recited by claim 4 is $f_{PB} = \{(1+C/Rcrown)\cdot\delta i+\delta o\}/\delta o$ (11), wherein, C is constant having a value in the range between 40 and 80. This maps to specification page 10, beginning with line 22.

The specific numeric values recited in the dependent claims are found in the claims as originally filed, specification pages 12-14.

A more detailed discussion of the invention follows in the Arguments section of this brief.

(vi) **Ground of Rejection to be Reviewed on Appeal**

Whether the rejection of claims 1-11 under §102(e) as anticipated by AKAGI et al. 6,612,954 ("AKAGI") was proper.

(vii) **Arguments**

General

The invention is a CVT drive belt (3) with a continuous band (11) having a radially inwardly oriented surface (12) and a radially outwardly oriented surface (13), and an array of transverse elements (20) engaging the continuous band (11) such that the elements (20) may slide along a longitudinal direction thereof.

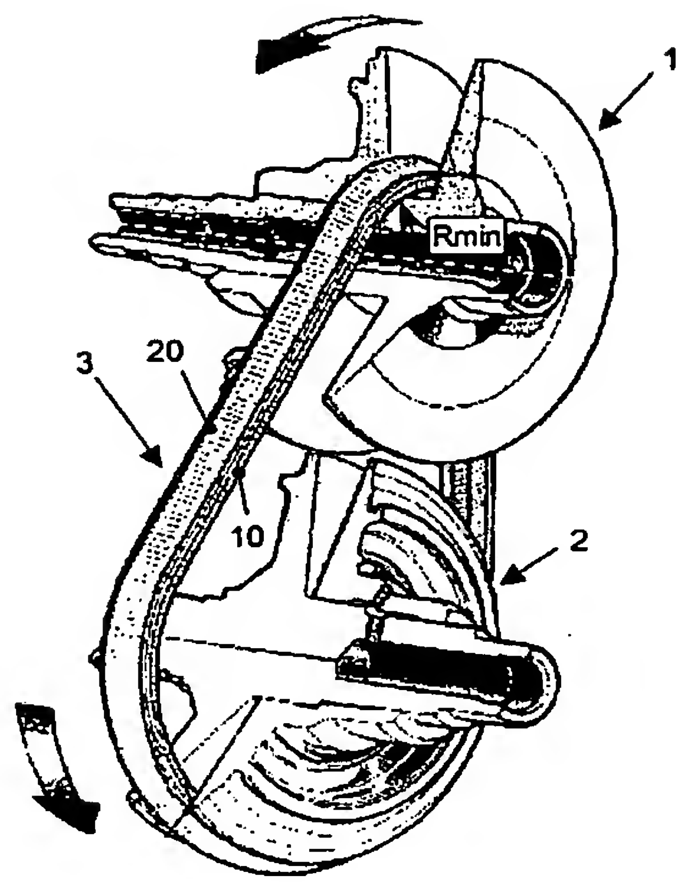


FIG. 1

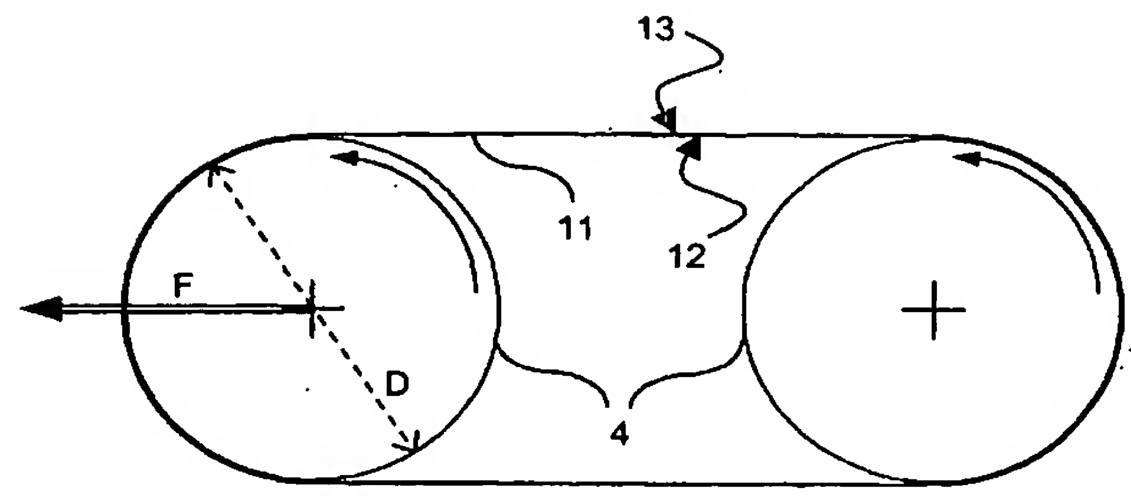


FIG. 4

Each independent claim requires the continuous band (11) be curved in a transverse direction at a non-infinite crowning radius of curvature Rcrown and provided with an internal residual stress distribution defining a curling radius of curvature Rcurl at which a continuous band (11) would be curved in its longitudinal direction when cut. Figure 6 depicts a section of a cut a continuous band drawn in perspective and defining the crowning radius and the curling radius.

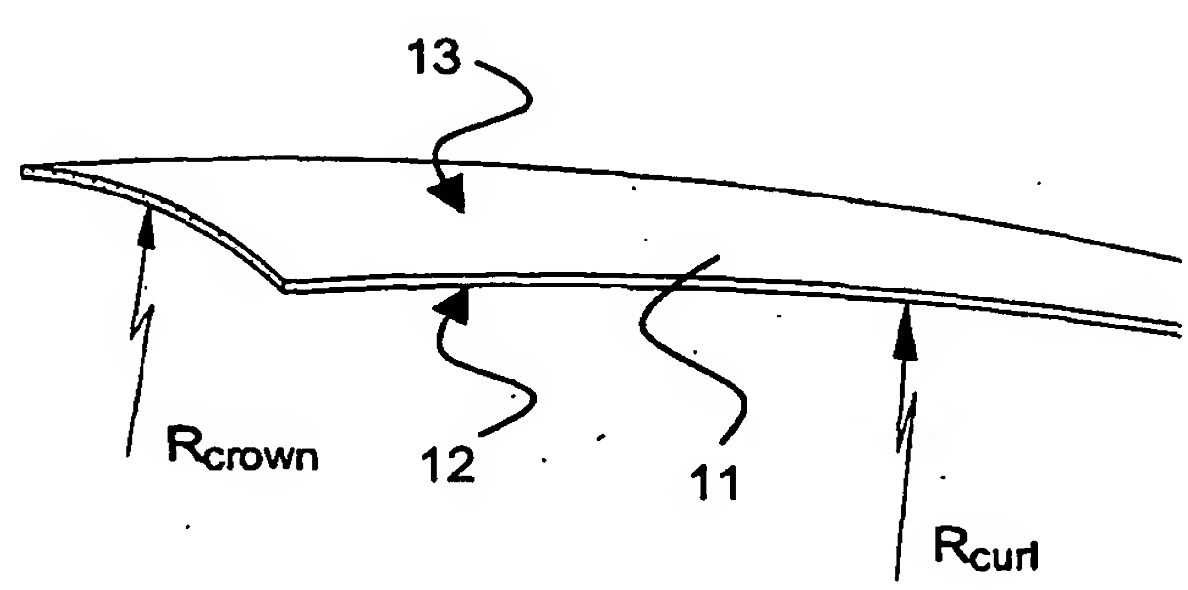


FIG. 6

During operation, the continuous band (11) can be bent in its longitudinal direction at a minimum radius of curvature Rmin and whereby a ratio between the curling radius and the minimum radius Rcurl/Rmin defines a pre-bending factor f_{PB}. The pre-bending factor f_{PB} satisfies various equations recited in claims 1, 4, and 7.

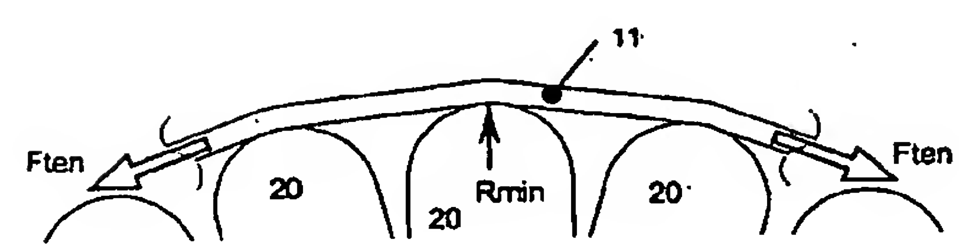


FIG. 3

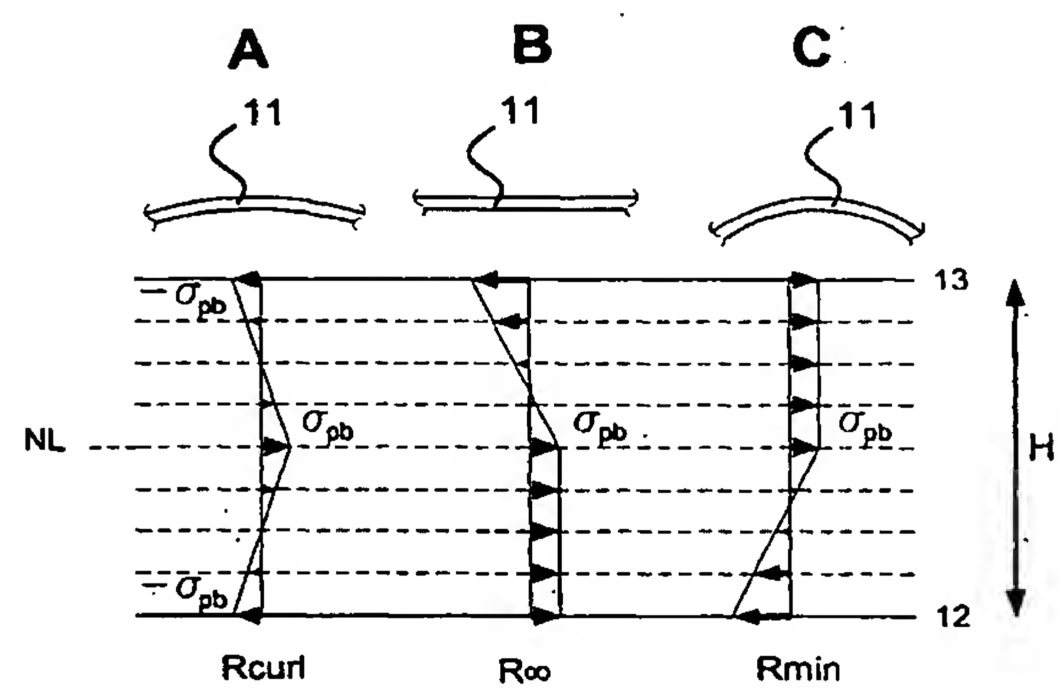


FIG. 5

Rmin is indicated in Figure 1 and in Figures 3 and 5. Figure 3 illustrates the influence of the polygon effect on the minimum radius of curvature Rmin at which the continuous band 11 is bent and subjected to a tensile force Ften in the longitudinal direction during operation. Because of the finite number of transverse elements 20 that support the band 11 when the drive belt 3 is in a longitudinally bent trajectory part as it runs between the discs of a pulley, the minimum radius of curvature Rmin may be somewhat smaller than the average radius of curvature of the belt 3, i.e., the radius of curvature of the band 11 when it would be smoothly

and continuously curved along the bent trajectory. The invention takes into account that locally the minimum radius of curvature Rmin of the continuous band is smaller than an average radius of curvature.

Figure 4 (above) schematically depicts the pre-bending process used to provide the band 11 with an internal residual stress distribution. In this pre-bending process, the band 11 is mounted around two rollers 4 that are simultaneously rotated and moved apart as indicated by the arrow F to such an extent that the band 11 plastically deforms. The diameter D of the rollers 4 thereby determines the distribution of the internal residual stress, which distribution is characterised by the radius of curvature in the longitudinal direction of the band 11 when it is cut perpendicular thereto, i.e. the curling radius Rcurl.

Figure 5 illustrates the internal stress distribution due to longitudinal bending in a side elevation of a section of the pre-bent continuous of the band 11, for three postures thereof that are characterised by their curvature or amount of bending in the longitudinal direction. σ_{PB} denotes the largest internal residual tension and compression stress introduced in the pre-bending process. The neutral line NL represents the line where the stress due to bending is zero. On opposite sides of this line, the said

stress has an opposite sign, i.e. changes from a tension stress to a compression stress.

The continuous band 11 curvature A corresponds to the curling radius Rcurl whereby the compression stress at the radially inwardly oriented surface 12 and the compression stress at the radially outwardly oriented surface 13 are equal and also relatively small. Curvature B shows the band 11 straightened (the section of the drive belt 3 that is located between the pulleys 1 and 2) and the inner fibre stress is a positive or tension stress and the outer fibre stress is a negative or compression stress. Curvature C shows the band 11 longitudinally bent at the minimum radius of curvature in the longitudinal direction Rmin that occurs during operation of the drive belt 3 in the transmission. In this posture, the outer fibre stress is a tension stress and the inner fibre stress is a compression stress, whereby the absolute outer fibre stress level is at an overall maximum level (specification page 6, lines 25 to page 7, line 4).

The inventors recognized (specification page 8, lines 7 *et seq)* that the crowning radius Rcrown that is provided to the bands 11 has a significant influence on the internal stress distribution as has the phenomenon of anticlastic bending. Figure 6 depicts a section of a continuous band 11 defining the crowning radius Rcrown and the

curling radius Rcurl. The invention defines an optimum design of the band 11 in terms of its curling radius Rcurl based thereon. The analysis according to the invention recognizes that the transverse curvature of continuous bands 11 that is nowadays applied significantly influences its longitudinal bending, and more in particular, the maximum separation between the inner and the outer fibres 12 and 13 increases as the crowning radius Rcrown of the band 11 decreases. Moreover, when the crowning radius Rcrown decreases, a maximum inner fibre distance δi that is measured between the inner fibre 12 and the neutral line NL for pure bending increases more than a maximum outer fibre distance δo that is measured between the outer fibre 13 and the neutral line NL, which effect is illustrated in figure 7 in an exaggerated manner.

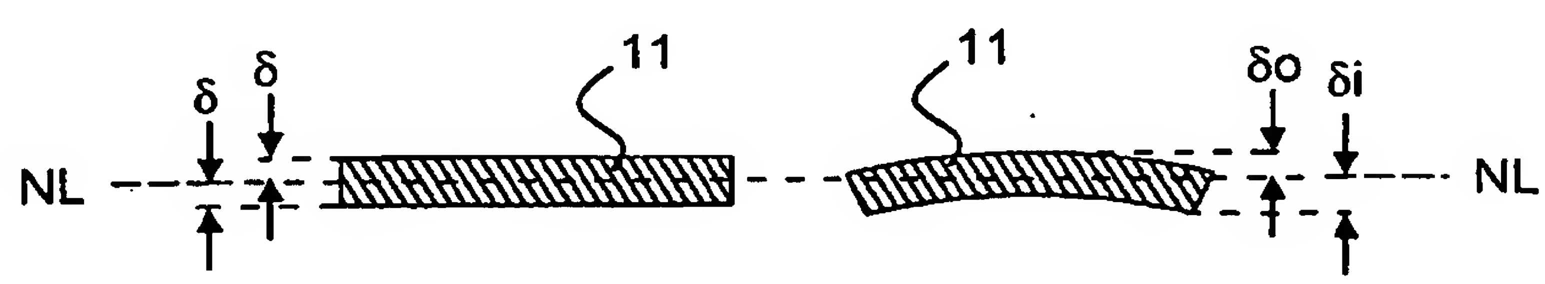


FIG. 7

The invention provides a result that is different from that previously considered the optimum value. Beginning at the top of specification page 8, reference is made to Figure 8 graphing the pre-bending factor f_{PB} according to the invention for the typically applied design of the continuous bands having a height of about 185 μm and a transverse width

of about 9.5 mm for a range of typically applied crowning radii Rcrown. As an example, the pre-bending factor f_{PB} is highlighted for a typically applied crowning radius of 100 mm when measured in a section of the continuous band 11 that is straightened and tensioned at $0.4 \cdot 10^3$ N, which pre-bending factor f_{PB} is approximately 2.3 and thus is substantially larger, about 15%, than what was previously considered the optimum value. Continuous bands satisfying the invention's pre-bending factor f_{PB} have an improved fatigue resistance.

The pre-bending factor recited by claim 4 is $f_{PB} = \{(1+C/Rcrown) \cdot \delta i+\delta o\}/\delta o$ (11), wherein, C is constant having a value in the range between 40 and 80.

The calculations in accordance with the invention showed that, at least for relatively small crowning radii Rcrown, equation (10) (recited by claim 7) can be approximated by: $f_{PB} = \{(1+C/Rcrown) \cdot \delta i+\delta o\}/\delta o$ (11) (recited by claim 4).

The invention results in a pre-bending factor that is considerably higher than the previously considered optimum value, i.e., 2.8 vs. 2.0, with validation tests showing a potential increase of up to 70% of drive belt service life (specification page 11, lines 16). The present invention uses a single, new criterion representing a combined, integral effect of these two known parameters (the band crowning radius and the band curling radius) to be taken into account for

optimizing tensile stress within a band. This approach results in a belt with a different structure, i.e., a different internal stress distribution.

Terms Used

It is noted that there are no formal matters/rejections outstanding. Nonetheless, appellants sense that, due to the complexity of the subject matter, there may be some unstated technical issues. However, the terms being used in the application, as well as the parameters being recited, are known to those of skill in the art.

The claims recite a prescribed relationship between the pre-bending factor and the crowning radius. That is to say, for a non-infinite, crowning radius Rcrown that is applied to the continuous band, a specific, optimum curling radius Rcurl is defined for the respective band in terms of the minimum radius of curvature at which the endless band is bent in its longitudinal direction during operation Rmin. The ratio of Rcurl/Rmin must satisfy a specified pre-bending factor equation.

All of the three radii recited have a distinct and commonly accepted meaning in the current area of technical expertise of the one of skill. Moreover, each such radius is sufficiently discussed in the specification as filed, or at least in the relevant prior art specifically cited for that

purpose in the specification. Apart from identifying the relevant parameters as such, the application also teaches how the respective parameters are measured and, in the case of Rcrown and Rcurl, how they can be controlled in the manufacturing process.

As for the means to be employed by one of skill for measuring an amount or radius of curvature of a curved surface (Rcrown of Rcurl), appellants consider this to be well within the ordinary skills. In fact, several methods and means are readily available as part of the common general knowledge.

Rcrown

The crowning radius Rcrown of the band is as such illustrated in Figure 6 of the application and is also defined in words on specification page 3, lines 27-33, where it is *inter alia* also referred to the state of art document EP-1 111 271 that is entirely devoted to the feature of the crowning radius and that also discloses a method for applying a desired crowning curvature to the band.

On page 9, lines 6-7, the present specification also mentions under what conditions the crowning radius should be measured.

Rcurl

The curling radius Rcurl of the band is as such illustrated in Figure 6 of the application and is also defined

in words on specification page 3, lines 19-21.

Further, on page 5, line 35 to page 6, line 8, the application discloses a method for applying a desired curling curvature to the band, which method is illustrated in Figure 4 and which method is known from the state of art document EP 0 279 645 specifically referred to in the application, which document is entirely devoted this parameter.

Finally, on specification page 2, lines 21-5 it is taught how the curling radius can be measured, including a reference to the state of art document EP 0 283 303 that is entirely devoted to such subject.

Rmin

The parameter, minimum radius of curvature Rmin of the drive belt during operation, appears to be self-explanatory, but is also employed in the calculations presented in the state of art document EP 0 279 645. Moreover, the present specification elucidates on this parameter on page 2, line 29 - page 3, line 8, as well as with reference to Figures 1 and 3.

Pre-bending factor f_{pb}

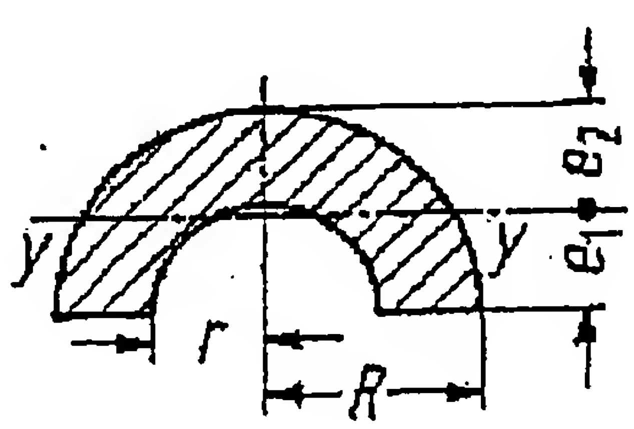
The pre-bending factor is determined as the ratio between Rcurl and Rmin.

The *preferred or prescribed* pre-bending factor is the specific ratio between Rcurl and Rmin that provides a

minimum internal stress in the bands during operation, i.e., providing the optimum curling radius Rcurl-opt for a given minimum radius Rmin (see derivation of eq. (5) in the application).

Claim 1 & 4: parameters δo and δi

The parameters of δo and δi employed in claims 1 and 4 are described on specification page 8, lines 20-26 with reference to Figure 7 and referring to pure bending, which feature and parameters themselves are a well established part of the common general knowledge. Pure bending of transversely curved (i.e. crowned) plate is a calculation of distance between neutral line/plane y and outer fibre (distance e_2 - δo and between neutral line/plane y and inner fibre (distance e_1 = δi) as follows:



$$I_y = 0{,}1098(R^4 - r^4) - 0{,}283 R^2 r^2 \frac{R-r}{R+r}$$

$$W_{y1,2} = I_y / e_{1,2}$$

$$\text{für } e_1 = \frac{4}{3\pi} \, \frac{R^2 + Rr + r^2}{R+r}$$

$$\text{bzw. } e_2 = R - e_1$$

Claim 7: parameters fo and fi

In relation to these latter parameters, see the disclosure and elucidation thereof in the application, page 9,

line 17 - page 10, line 21 with reference to Figure 9.

Also attached is a paper "Stress reduction in push belt rings using residual stresses" of the inventors, which was published at a conference only after filing of the present application. The paper illustrates the practical implications and significance of the present invention in sections 5.2 and 5.3 thereof in relation to the conventional considerations discussed in chapter 3 thereof. This paper was provided with the Amendment of February 13, 2006.

Again, although there are no formal matters outstanding, appellants sense that there may be some unstated technical issues. The above is provided to review that the terms being used in the application, as well as the parameters being recited, are known to those of skill in the art.

<u>Claim 1</u>

The Examiner asserts in the paragraph spanning pages 2-3 of the Official Action, that AKAGI discloses a drive belt (15) with a continuous band (31), "whereby during operation the continuous band (31) can be bent in its longitudinal direction at a minimum radius of curvature Rmin and whereby a ratio between the curling radius and the minimum radius Rcurl/Rmin defines a pre-bending factor f_{PB}, characterised in that, the pre-bending factor f_{PB} satisfies the equation: $f_{PB} = (\delta i+\delta o)/\delta o$ wherein: δi is the largest perpendicular distance

in the radial direction between a neutral line NL in the cross section of the continuous band where the stress due to pure longitudinal bending would be zero and the radially inner most surface of the band and δo is the largest perpendicular distance in the radial direction between the said neutral line NL and the radially outer most surface of the band (inherent)".

In making this assertion, other than identifying the continuous band 31 of AKAGI drive belt 15, the Examiner merely repeats the concluding recitations of claim 1 and provides the single word explanation "inherent".

To be fair, the Examiner has been co-operative with the undersigned attorney and discussed this case on plural occasions. However, the Examiner's position as to this case seems to be summarized in the "Response to Arguments" section found on Official Action page 3.

In this section, the Examiner has acknowledged some arguments put forth as to why AKAGI fails to anticipate. However, rather than to show where AKAGI discloses the recited features of the invention, or explain why those recited features would be inherent, the Examiner only states that "Applicant has failed to provide evidence that the claimed invention is structurally different from the cited prior art."

With respect, appellants are entitled to a patent unless the claims are shown to be anticipated or rendered obvious. The Examiner has not satisfied this burden. Therefore the rejection is improper.

Additionally, as discussed in the Summary of the Claimed Subject Matter portion of this brief, the pre-bending factor f_{PB} provided under the invention is greater than that of the prior art. Thus, the invention does provide bands that are structurally different from the prior art in that the inventive bands have a different internal stress distribution that results in, and can be demonstrated and measured by, the band being curved at the recited curling radius when it is cut. The internal stress distribution, and thus the curling radius, is determined (and controllable) in the manufacturing process of the band. Also the crowning radius is determined and controllable in the manufacturing process of the band. Thus, both the radii can be set and controlled independently from each other during band manufacture.

Appellants acknowledge that it is widely known that minimizing tensile stress within a band element of a metal belt is a known factor for optimizing transmittable force by a belt (in particular with respect to operating conditions of the belt). Further, appellants acknowledge that band crowning radius and band curling radius (band curling radius being an

effect of pre-bending) are also known.

However, these individually are not what is recited. What is recited is satisfying a specific pre-bending factor equation, .i.e., in claim 1: $f_{PB} = \{(f_i/f_o) \bullet \delta i + \delta o\} / \delta o$.

AKAGI deals only with the cross-wise stress distribution in the band as a consequence of the band crowning radius only. However, AKAGI does not consider the curling radius, nor suggests any combined effect of the crowning radius and the curling radius. Therefore, there is no reason to believe AKAGI would anticipate the pre-bending factor equation recited by claim 1.

AKAGI makes no teaching as to the concept of defining the known curling radius in terms of a neutral line and the criteria δi and δo defined in the claim 1 (that are used to quantify the band crowning radius). Therefore, AKAGI makes to teaching as to the underlying insight provided by the present application and AKAGI cannot serve as the basis to achieve the recited optimization. Therefore, AKAGI cannot have the recited structure.

There is no support for AKAGI **inherently** satisfying the recitations of claim 1.

In relation to claim 1, one cannot assume that AKAGI exhibits the recited curling radius and pre-bending factor defined as f_{PB} therein.

Nor is it necessary that AKAGI have the recited curling radius and pre-bending factor. This cannot be the case, since the parameters of band curling radius and band crowning radius can be set and are controllable independently from one another in the manufacturing process. Accordingly, the recited curling radius and pre-bending factor are not necessarily inherent features of the crowning radius.

Although AKAGI discloses crowning radius, it is especially noteworthy that AKAGI does not even mention the curling radius or the pre-bending factor. Only the present invention teaches to take into account the value of the crowning radius that is provided to the band (i.e., set in the manufacturing process thereof) in determining and subsequently setting the value of the curling radius (defined in terms of the pre-bending factor). This results in the curling radius being independently provided to the band in the manufacturing process.

AKAGI cannot be said to inherently satisfy the recited pre-bending factor since the parameters of crowning radius and curling radius are structurally unrelated and can in fact be set independently in the manufacturing process.

Further, the Examiner has the burden in establishing inherency. Reference is made to the MPEP, parts reproduced below (emphasis added).

2112 Requirements of Rejection Based on Inherency; Burden of Proof [R-3]

The express, implicit, and inherent disclosures of a prior art reference may be relied upon in the rejection of claims under **35 U.S.C. 102** or **103**. "The inherent teaching of a prior art reference, a question of fact, arises both in the context of anticipation and obviousness." *In re Napier*, 55 F.3d 610, 613, 34 USPQ2d 1782, 1784 (Fed. Cir. 1995) (affirmed a **35 U.S.C. 103** rejection based in part on inherent disclosure in one of the references). See also *In re Grasselli*, 713 F.2d 731, 739, 218 USPQ 769, 775 (Fed. Cir. 1983).

IV. EXAMINER MUST PROVIDE RATIONALE OR EVIDENCE TENDING TO SHOW INHERENCY

The fact that a certain result or characteristic may occur or be present in the prior art is not sufficient to establish the inherency of that result or characteristic. *In re Rijckaert*, 9 F.3d 1531, 1534, 28 USPQ2d 1955, 1957 (Fed. Cir. 1993) (reversed rejection because inherency was based on what would result due to optimization of conditions, not what was necessarily present in the prior art); *In re Oelrich*, 666 F.2d 578, 581-82, 212 USPQ 323, 326 (CCPA 1981). **"To establish inherency, the extrinsic evidence 'must make clear that the missing descriptive matter is necessarily present in the thing described in the reference, and that it would be so recognized by persons of ordinary skill. Inherency, however, may not be established by probabilities or possibilities. The mere fact that a certain thing may result from a given set of circumstances is not sufficient.' "** *In re Robertson*, 169 F.3d 743, 745, 49 USPQ2d 1949, 1950-51 (Fed. Cir. 1999) (citations omitted) (The claims were drawn to a disposable diaper having three fastening elements. The reference disclosed two fastening elements that could perform the same function as the three fastening elements in the claims. The court construed the claims to require three separate elements and held that the reference did not disclose a separate third fastening element, either expressly or inherently.). >Also, "[a]n invitation to investigate is not an inherent disclosure" where a prior art reference "discloses no more than a broad genus of potential applications of its discoveries." *Metabolite Labs., Inc. v. Lab. Corp. of Am. Holdings*, 370 F.3d 1354, 1367, 71 USPQ2d 1081, 1091 (Fed. Cir. 2004) (explaining that "[a] prior art reference that discloses a genus still does not inherently disclose all species within that broad category" but must be examined to see if a disclosure of the claimed species has been made or whether the prior art reference merely invites further experimentation to find the species.<

The MPEP makes clear that an inherency rejection cannot be made on mere speculation. Indeed, appellants respectfully submit that the absence within the prior art of the necessary teachings to achieve the recited curling radius

and pre-bending factor make clear that AKAGI does not inherently disclose the recited features of the present invention.

The Examiner has failed to establish inherency by failing to provide extrinsic evidence making clear that the missing descriptive matter (the pre-bending factor equation being satisfied) is necessarily present in the thing described in the reference (the continuous band of AKAGI), and that it would be so recognized by persons of ordinary skill. Indeed, the Examiner has also failed to even establish any case concerning the probability of the recited equation being satisfied. Even so and as noted in the above MPEP section, inherency, however, may not be established by probabilities or possibilities. The mere fact that a certain thing may result from a given set of circumstances is not sufficient.

There being no showing the AKAGI discloses a structure that satisfies the recited pre-bending factor equation, the anticipation rejection is improper.

Claim 4

The arguments as to claim 1 also apply to claim 4.

Claim 4 requires satisfying the equation $f_{PB} = \{(1+C/Rcrown)\bullet\delta i+\delta o\}/\delta o$ (11) wherein: - C is constant having a value in the range between 40 and 80.

The Examiner does not assert AKAGI meets this recitation.

The Examiner makes no proof as to AKAGI meeting this recitation.

There being no showing the AKAGI discloses a structure that satisfies the recited pre-bending factor equation, the anticipation rejection is improper.

Claim 7

The arguments as to claim 1 also apply to claim 7.

Claim 7 requires satisfying the equation $f_{PB} = \{(f_i/f_o)\cdot\delta i+\delta o\}/\delta o$ wherein: - f_i is a stress factor defining the relative increase of the maximum tension stress at the radially inner most surface (12) due to anticlastic bending when the band (11) is bent straight, and - f_o is a stress factor defining the relative increase of the maximum tension stress at the radially outermost surface (13) due to anticlastic bending when the band (11) is longitudinally curved at the said minimum radius of curvature Rmin.

The Examiner does not assert AKAGI meets this recitation.

The Examiner makes no proof as to AKAGI meeting this recitation.

There being no showing the AKAGI discloses a structure that satisfies the recited pre-bending factor

equation, the anticipation rejection is improper.

Dependent claims

Claim 2, 5, and 11

The arguments as to claim 1 also apply to claims 2, 5 and 11.

Claim 2 recites that the radius of curvature Rcrown of the continuous band (11) in the transverse direction, when measured as the band (11) is straightened and tensioned in the longitudinal direction, has a value in the range between 50 mm and 1000 mm; claims 5 and 11 recite having a value in the range between 50 mm and 250 mm.

The Examiner does not assert AKAGI meets these recitations. The Examiner makes no proof as to AKAGI meeting these recitations.

There being no showing the AKAGI discloses a structure that satisfies the recited features, the anticipation rejection is improper.

Claim 3

The arguments as to claim 1 also apply to claim 3.

Claim 3 requires that the pre-bending factor f_{PB} has a value in the range between 2.15 and 2.45.

The Examiner does not assert AKAGI meets this recitation. The Examiner makes no proof as to AKAGI meeting this recitation.

There being no showing the AKAGI discloses a structure that satisfies the recited pre-bending factor equation, the anticipation rejection is improper.

Claim 6

The arguments as to claim 1 also apply to claim 6.

Claim 6 requires that the pre-bending factor f_{PB} has a value in the range between 2.40 and 3.60.

The Examiner does not assert AKAGI meets this recitation. The Examiner makes no proof as to AKAGI meeting this recitation.

There being no showing the AKAGI discloses a structure that satisfies the recited pre-bending factor equation, the anticipation rejection is improper.

Claims 8, 9, and 10

These claims recite a continuously variable transmission having a drive belt as recited by claims 7, 1, and 4 respectively. These claims stand or fall with each of claims 7, 1, and 4 respectively.

The anticipation rejection as to each claim is improper. Reversal of the anticipation rejection as to each claim is accordingly respectfully requested.

Respectfully submitted,

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REL/fb

(viii) **Claims Appendix**

1. Drive belt (3) for a transmission realising a continuously variable transmission ratio comprising at least one continuous band (11), having a radially inwardly oriented surface (12) and a radially outwardly oriented surface (13), and an array of plate-like transverse elements (20) engaging said continuous band (11) such that the elements (20) may slide along a longitudinal direction thereof, which continuous band (11) is curved in a transverse direction at a non-infinite crowning radius of curvature Rcrown and is provided with an internal residual stress distribution defining a curling radius of curvature Rcurl at which a continuous band (11) would be curved in its longitudinal direction when cut, whereby during operation the continuous band (11) can be bent in its longitudinal direction at a minimum radius of curvature Rmin and whereby a ratio between the curling radius and the minimum radius Rcurl/Rmin defines a pre-bending factor f_{PB}, characterised in that, the pre-bending factor f_{PB} satisfies the equation:

$$f_{PB} = (\delta i+\delta o) / \delta o$$

wherein:

- δi is the largest perpendicular distance in the radial direction between a neutral line NL in the cross section of the continuous band (11) where the stress due to

pure longitudinal bending would be zero and the radially inner most surface (12) of the band (11) and

- δo is the largest perpendicular distance in the radial direction between the said neutral line NL and the radially outer most surface (13) of the band (11).

2. Drive belt (3) according to claim 1, characterised in that the radius of curvature Rcrown of the continuous band (11) in the transverse direction, when measured as the band (11) is straightened and tensioned in the longitudinal direction, has a value in the range between 50 mm and 1000 mm.

3. Drive belt (3) according to claim 2, characterised in that the pre-bending factor f_{PB} has a value in the range between 2.15 and 2.45.

4. Drive belt (3) for a transmission realising a continuously variable transmission ratio comprising at least one continuous band (11), having a radially inwardly oriented surface (12) and a radially outwardly oriented surface (13), and an array of plate-like transverse elements (20) engaging said continuous band (11) such that the elements (20) may slide along a longitudinal direction thereof, which continuous

band (11) is curved in a transverse direction at a non-infinite crowning radius of curvature Rcrown and is provided with an internal residual stress distribution defining a curling radius of curvature Rcurl at which a continuous band (11) would be curved in its longitudinal direction when cut, whereby the continuous band (11) can be bent in its longitudinal direction at a minimum radius of curvature Rmin and whereby a ratio between the curling radius and the minimum radius Rcurl/Rmin defines a pre-bending factor f_{PB}, characterised in that, the pre-bending factor f_{PB} satisfies the equation:

$$f_{PB} = \{(1+C/Rcrown)\bullet\delta i+\delta o\}/\delta o \quad (11)$$

wherein:

- C is constant having a value in the range between 40 and 80,

- δi is the largest perpendicular distance in the radial direction between a neutral line NL in the cross section of the continuous band (11) where the stress due to pure longitudinal bending would be zero and the radially inner most surface (12) of the band (11), and

- δo is the largest perpendicular distance in the radial direction between the said neutral line NL and the radially outer most surface (13) of the band (11).

5. Drive belt (3) according to claim 4, characterised in that the radius of curvature Rcrown of the continuous band (11) in the transverse direction, when measured as the band (11) is straightened and tensioned in the longitudinal direction, has a value in the range between 50 mm and 250 mm.

6. Drive belt (3) according to claim 5, characterised in that the pre-bending factor f_{PB} has a value in the range between 2.40 and 3.60.

7. Drive belt (3) for a transmission realising a continuously variable transmission ratio comprising at least one continuous band (11), having a radially inwardly oriented surface (12) and a radially outwardly oriented surface (13), and an array of plate-like transverse elements (20) engaging said continuous band (11) such that the elements (20) may slide along a longitudinal direction thereof, which continuous band (11) is curved in a transverse direction at a non-infinite crowning radius of curvature Rcrown and is provided with an internal residual stress distribution defining a curling radius of curvature Rcurl at which a continuous band (11) would be curved in its longitudinal direction when cut, whereby the continuous band (11) can be bent in its

longitudinal direction at a minimum radius of curvature Rmin and whereby a ratio between the curling radius and the minimum radius Rcurl/Rmin defines a pre-bending factor f_{PB}, characterised in that, the pre-bending factor f_{PB} satisfies the equation:

$$f_{PB} = \{(f_i/f_o)\bullet\delta i+\delta o\}/\delta o$$

wherein:

- f_i is a stress factor defining the relative increase of the maximum tension stress at the radially inner most surface (12) due to anticlastic bending when the band (11) is bent straight,

- f_o is a stress factor defining the relative increase of the maximum tension stress at the radially outermost surface (13) due to anticlastic bending when the band (11) is longitudinally curved at the said minimum radius of curvature Rmin,

- δi is the largest perpendicular distance in the radial direction between a neutral line NL in the cross section of the continuous band (11) where the stress due to pure longitudinal bending would be zero and the radially inner most surface (12) of the band (11), and

- δo is the largest perpendicular distance in the radial direction between the said neutral line NL and the radially outer most surface (13) of the band (11).

8. Continuously variable transmission comprising a drive belt (3) according to claim 7 and two pulleys (1, 2) that each define a tapered and substantially torus-shaped groove of variable width, in which groove a longitudinally curved section of a drive belt 3 is mounted, whereby during operation of the transmission the said section is bent at a smallest radius of curvature in the longitudinal direction Rmin.

9. Continuously variable transmission comprising a drive belt (3) according to claim 1 and two pulleys (1, 2) that each define a tapered and substantially torus-shaped groove of variable width, in which groove a longitudinally curved section of a drive belt 3 is mounted, whereby during operation of the transmission the said section is bent at a smallest radius of curvature in the longitudinal direction Rmin.

10. Continuously variable transmission comprising a drive belt (3) according to claim 4 and two pulleys (1, 2) that each define a tapered and substantially torus-shaped groove of variable width, in which groove a longitudinally curved section of a drive belt 3 is mounted, whereby during

operation of the transmission the said section is bent at a smallest radius of curvature in the longitudinal direction Rmin.

11. Drive belt (3) according to claim 1, characterised in that the radius of curvature Rcrown of the continuous band (11) in the transverse direction, when measured as the band (11) is straightened and tensioned in the longitudinal direction, has a value in the range between 50 mm and 250 mm.

(ix) **Evidence Appendix**

- Paper entitled "Stress reduction in push belt rings using residual stresses". As noted above, this paper was provided with the Amendment of February 13, 2006. This paper was entered, as a matter of right, in the record by the Examiner by the Official Action of May 18, 2006, which Official Action was responsive to the Amendment of February 13, 2006.

(x) **Related Proceedings Appendix**

None.

Stress reduction in push belt rings using residual stresses

An approach towards increased power density for push belt CVT's

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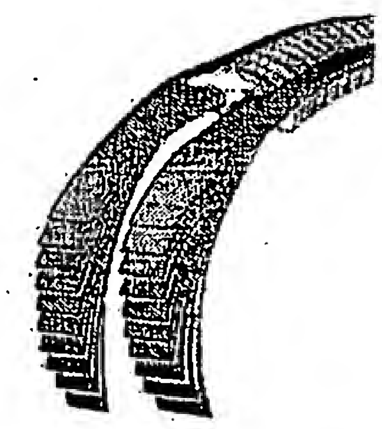


Summary

Since the push belt CVT first came into production, customer specifications on transmittable power, torque, space envelope, ratio coverage and durability have been extended. In answer to these changing demands, Van Doorne's Transmissie (VDT) dedicates itself to a continuous effort to improve the power density of its push belt.

Power density can be increased by reduction of critical stress levels in the rings of the belt. Within the current belt interface this is realised by improved pre-bending.
Rings are pre-bent in production. The process introduces a residual stress profile in the ring that helps to lower stress levels in critical areas during operation in the variator. Ring load analysis indicates that further improvement of the pre-bend process is possible.

This paper describes a theoretical model that helps to understand the stress critical areas in the rings that can be influenced by the pre-bend process. Theory is checked with experimental results revealing that model and test are consistent. Improvements have been implemented in the new push belt design to increase the power density of CVT applications.

1. Introduction

Over the last years the CVT market has seen a sharp increase in the number of applications. For the year 2002 numbers are foreseen to double. At this moment most car manufacturers have a car with CVT in their program or are working to get there.

For VDT, the main supplier of steel push belts, this increase in demand means a tremendous effort. In order to keep pace, large steps in production numbers are scheduled leading to a production increase of 590,000 units in the year 2001 towards 1,200,000 units in the year 2002. Together with the increase in numbers also an increase in application range is noticed. To reduce the level in transmission diversity, customer requirements regarding transmittable power, torque, space envelope, ratio coverage and durability are becoming more severe. In meeting those requirements, power density of the belt needs to be extended beyond the current state of the art belt design. To enable this at short notice, solutions have to be found within the current belt interface.

Power density of a steel push belt mainly is determined by the fatigue limit of the ring material [1]. To secure belt durability, ring stress must not exceed a certain level. An increase of power density without a ring stress increase can be achieved by changes in geometry.

Next to pure tension, bending plays an important role in the sum of stresses working on the rings. Bending stress levels can form up to 40% of the stress level induced during maximum load situations in the variator. They result from the radii the belt encounters in the variator and therefore are largest at the minimum running radii in the ratios Low and Overdrive.

Stresses from bending can also be brought into the ring during production before it is subjected to loads in the variator. This so-called residual stress, resulting from plastic deformation of the ring, can help to reduce the stress levels in critical areas during operation in the variator.

2. Introduction of residual stresses in push belt rings during production

After the basic ring is formed during the successive production steps pipe-rolling, welding, pipe-slitting, deburring and rolling, all previously introduced stresses are removed by annealing the ring. Then residual stresses are brought into the ring by means of three processes:

1. Calibration or pre-bend process : Brings the ring to a correct (calibration) length.
2. Hardening process : Increases the yield and fatigue limit of the material.
3. Nitriding process : Increases the wear resistance of the ring surfaces.

All processes can be optimised by adjustment of process parameters. This paper focuses on the optimisation of the pre-bend process of which **figure 1** shows a schematic view.

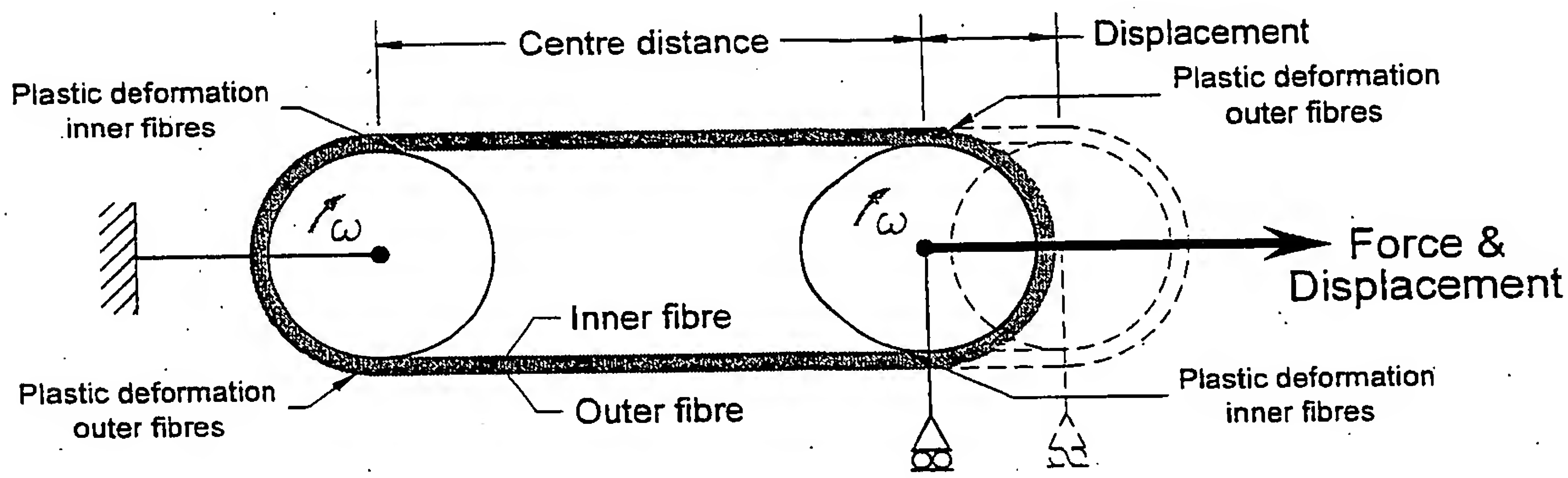


Fig. 1: Calibration or pre-bend process: calibration length definition and mechanical residual stress introduction.

During the pre-bend process the ring is elongated to a correct length. At the straight parts the ring is under tension. At the rollers the ring is bent and an additional stress is introduced. During a centre distance increase of the rollers the ring material is stressed beyond its yield limit and deformation becomes plastic.

Plastic deformation starts at the surface fibres when the ring goes from straight to roller radius and vice versa. During ring elongation, plastic deformation runs up to the neutral line. The stress profiles based on ideal plastic material behaviour are shown in **figure 2**.

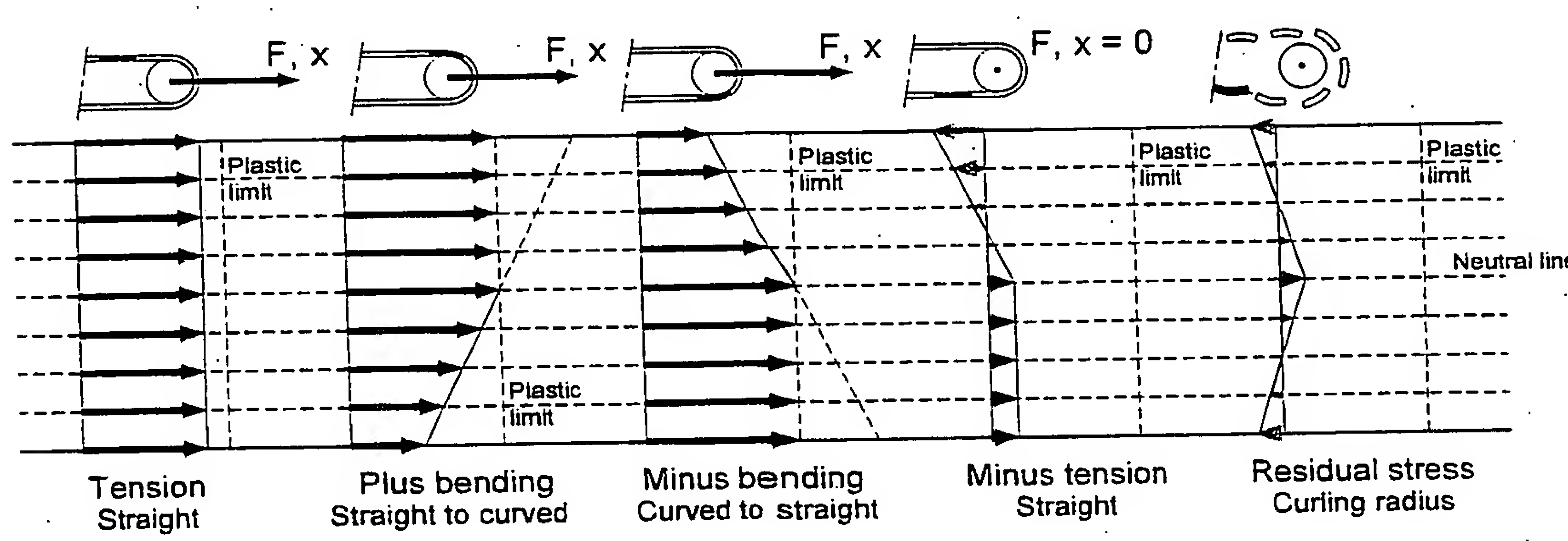


Fig. 2: Ring stress profiles during the pre-bend process. Input for the process is the annealed (stress-free) ring.

After the process a bending moment is left in the ring. This can be seen in case a ring part is cut free from the ring and curls up to a certain radius called the curling radius. In this situation the resulting bending moment from residual stresses on the ring part is zero and the residual stress profile resembles the last stress profile in figure 2. The cross-section contains compressive stress at the inner and outer fibre and tension stress at the neutral line.
In case the ring is bent straight the required bending moment leads to the fore last stress profile in figure 2.

3. Optimum residual stress profile

During operation in the variator the ring with curling radius is subjected to stresses from bending. Most critical load cases concerning bending are:

1. Bending the ring straight causing maximum tension at the inner ring fibres.
2. Bending the ring to its minimum running radius causing maximum tension at the outer ring fibres.

Figure 3 shows these load cases starting with the curling radius containing residual stresses from pre-bending. Only stresses from bending the ring are reviewed.

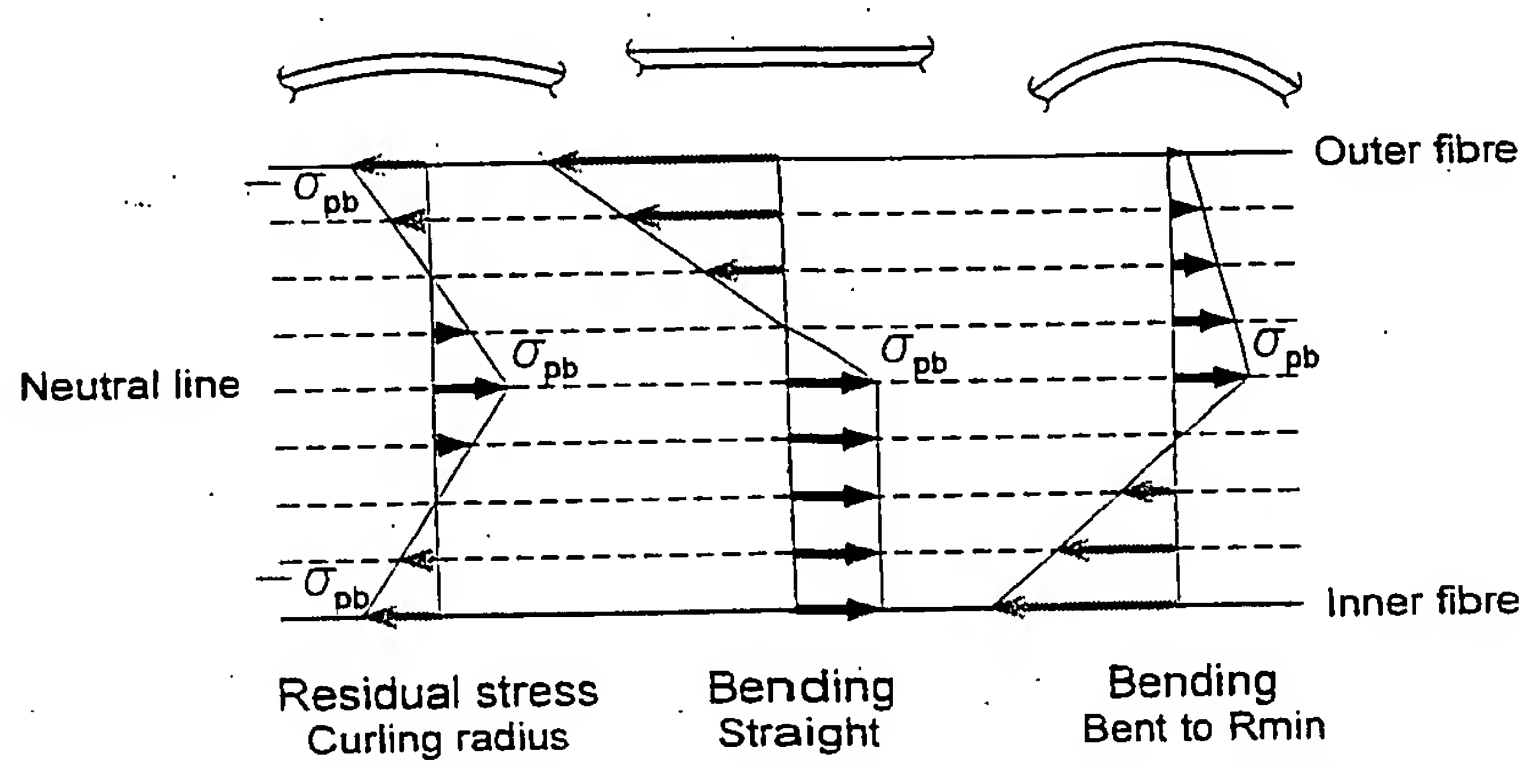


Fig. 3: Ring stress profiles in variator during critical load cases. Not optimal situation, only bending included. Hardening/nitriding excluded.

The stress profiles have maximum values at the inner, outer and neutral fibre. The tension stress at the neutral line is not influenced by bending and thus is determined by the pre-bend process. It is the maximum level of residual stress denoted by σ_{pb}.

Bending of straight beams can be described using simple beam theory. Maximum stress in longitudinal direction σ and bending moment M are calculated using the following equations:

$$\sigma = \frac{\delta \cdot E}{R}$$

(1)

$$M = \frac{E \cdot I}{R}$$

Where:

δ = Maximum fibre distance from neutral line [mm]

R = Bending radius neutral line [mm]

E = Young's modulus of elasticity [N/mm^2]

I = Moment of inertia beam cross-section [mm^4]

For curved rings, equations (1) can be used as an approximation. The curling and bending radii of the rings are sufficiently large compared to their maximum fibre distance to allow this. Stress deviations lie below 0.2%. A more accurate calculation method is found in [2].

The curling radius R_{curl} determines the stress profile in a straightened ring part. To bend the ring straight, a bending moment applying a maximum stress level σ_{bs} is required:

$$\sigma_{bs} = \frac{\delta \cdot E}{R_{curl}} \tag{2}$$

At the minimum running radius for the ring in the variator, denoted by R_{min}, the ring is subjected to a bending moment in the opposite direction. The maximum stress σ_{bmax} occurring while bending an initially straight ring part to this minimum running radius is:

$$\sigma_{b\max} = \frac{\delta \cdot E}{R_{\min}} \tag{3}$$

For tension stress minimisation it is desirable that maximum tension stress at the inner fibre (at straight part) equals maximum tension stress at the outer fibre (at minimum running radius). This means:

$$-\sigma_{pb} + \sigma_{bs} = -\sigma_{pb} - \sigma_{bs} + \sigma_{b\max} \tag{4}$$

Or:

$$-\sigma_{pb} + \frac{\delta \cdot E}{R_{curl}} = -\sigma_{pb} - \frac{\delta \cdot E}{R_{curl}} + \frac{\delta \cdot E}{R_{min}} \tag{5}$$

Which leads to:

$$R_{curl} = f_{pb} \cdot R_{min} \qquad \text{where,} \quad f_{pb} = 2 \ [-] \tag{6}$$

The theoretical optimal pre-bend factor $f_{pb} = 2$ is the basis for the pre-bend process. For rings with optimum curling radius, stress profiles now look like **figure 4**. Both bending the ring straight and bending the ring to the minimum running radius lead to identical but mirrored stress profiles.

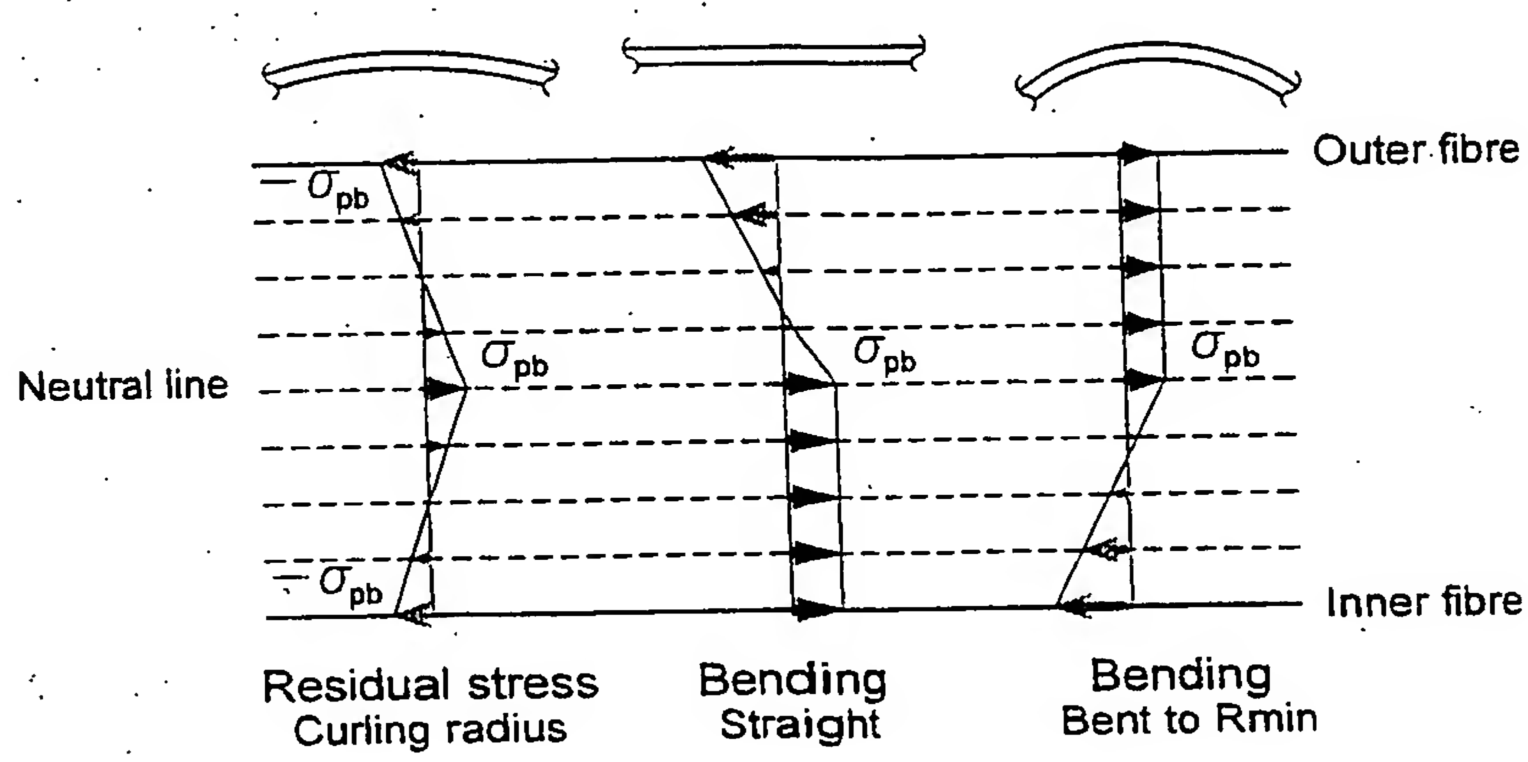


Fig. 4: Ring stress profiles in variator during critical load cases. Optimal situation, only bending included. Hardening/nitriding excluded.

4. Influences from subsequent production processes

After pre-bending, the hardening and nitriding process influence the residual stress profile in the ring leading to a different curling radius than obtained from the pre-bend process. It is shown equation (6) remains valid as long as the rings resulting from these processes answer to the optimum curling radius.

Hardening and nitriding cause relaxation in the ring material. During relaxation, the initial residual stress level in the material decreases with a certain factor. This factor depends on process time, process temperature and initial residual stress as shown in **figure 5**.

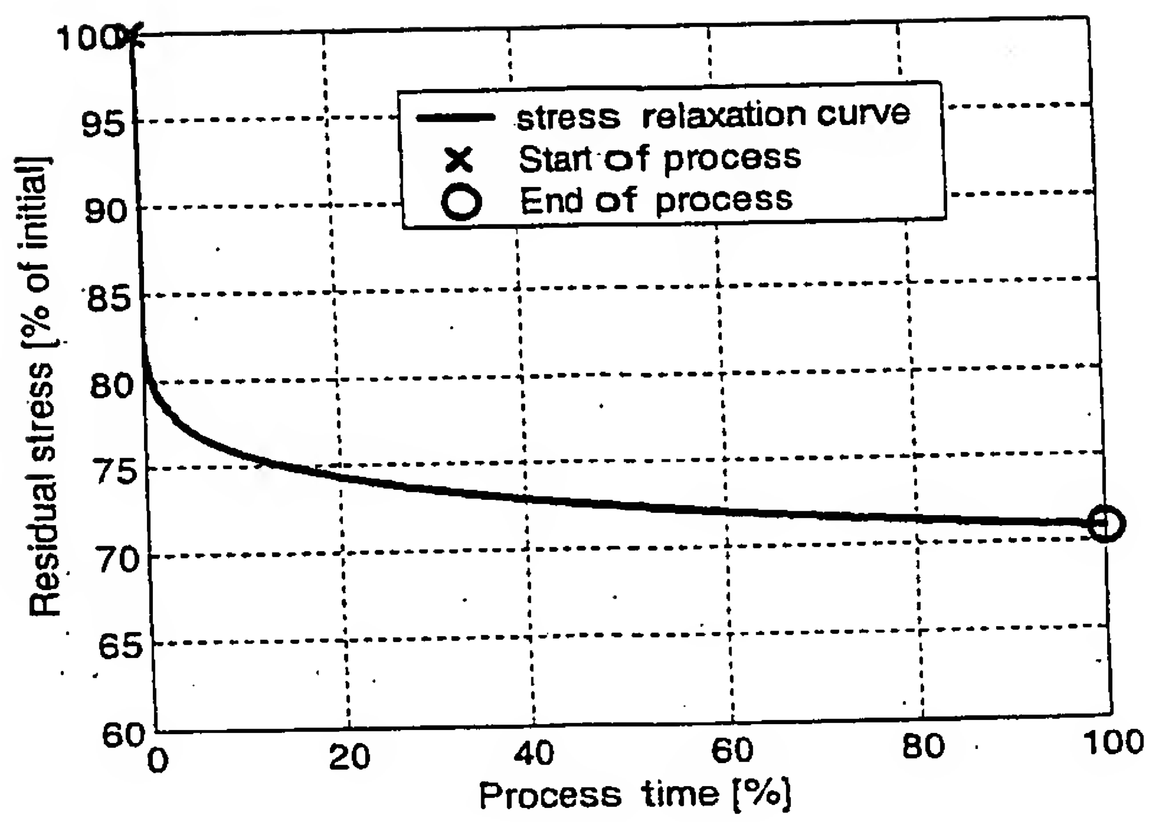


Fig. 5: Residual stress relaxation during a (constant) high temperature process (example from literature [3]).

As process time and temperature preferably are equal for all rings, the final factor mainly is a function of the initial residual stress level introduced by pre-bending.

During nitriding, compression stress is brought into a thin surface layer of the material. This stress is balanced by a tension stress working on the material in the centre as shown in **figure 6**.

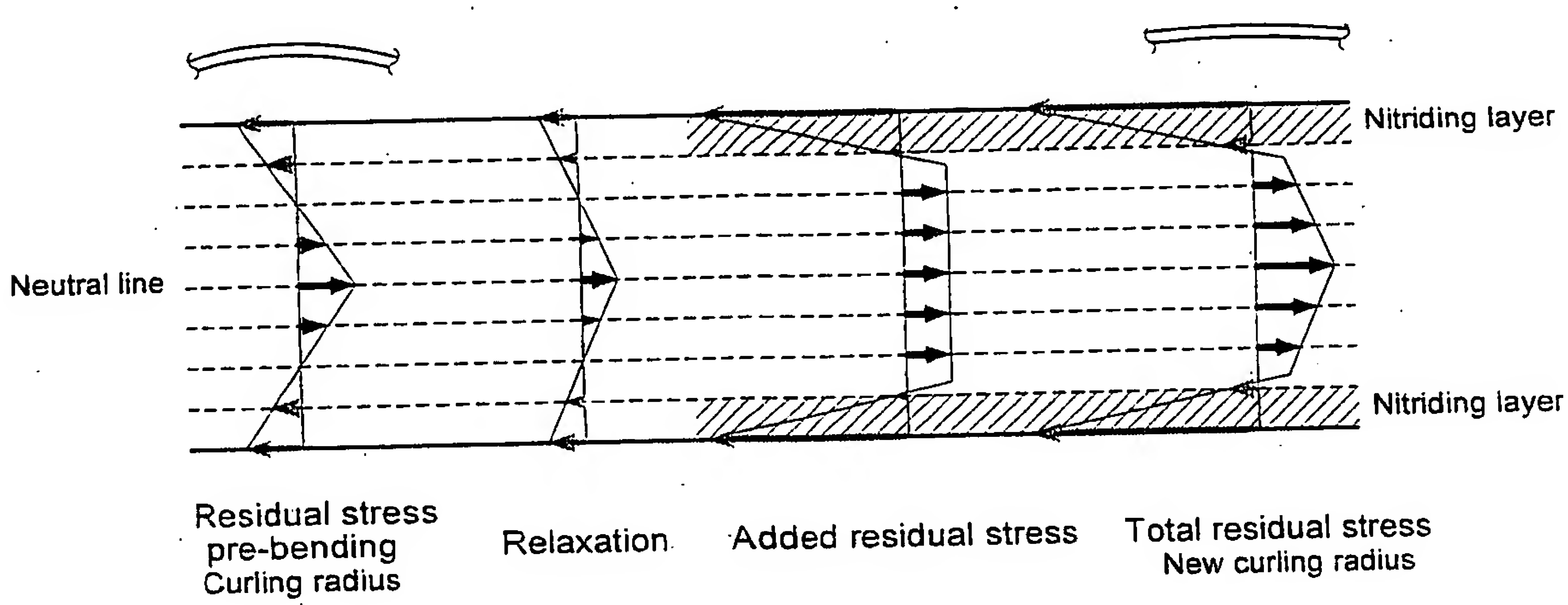


Fig. 6: Influences hardening and nitriding process on residual stress for a free ring part.

The relaxation of mechanical residual stress during hardening and nitriding causes curling radii originating from pre-bending to increase.

The symmetrical residual stress profile added by nitriding does not influence equation (6). It falls out of equation (4) because of its symmetry.

The maximum ring stress level occurring during operation in the variator is influenced by relaxation. Optimal combinations of pre-bending and relaxation result in optimal curling radii. Consequently the bending moments on the ring for bending straight and bending to the minimum radius cause identical maximum stress levels past the nitriding layer for both sides of the ring. This optimal situation is shown in **figure 7**. Stresses at the surface remain compressive as a result from the nitriding layer. The largest tension stresses occur between the nitriding layers.

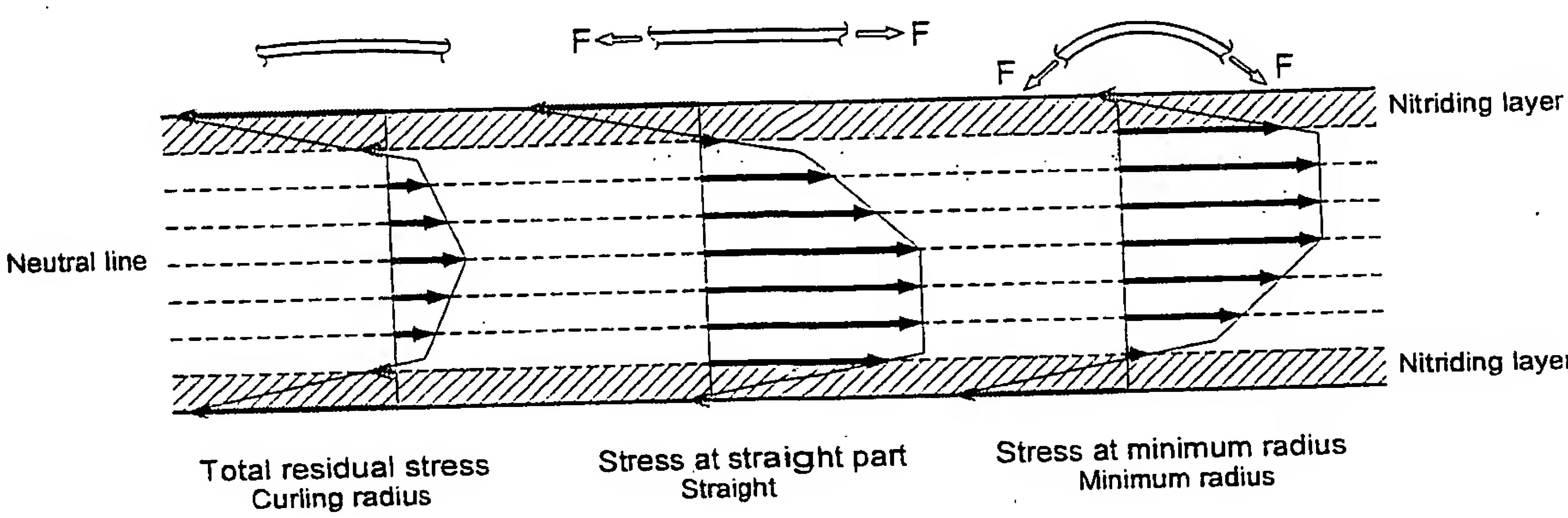


Fig. 7: Ring stress profiles in the variator, bending and tension included. Hardening/nitriding included.

For present commercial belt applications, the production settings for pre-bending, hardening and nitriding aim to generate rings of which the final curling radii answer to the optimum pre-bend factor $f_{pb} = 2$. In this case the bending moment required for bending the hardened and nitrided ring straight equals the bending moment for bending the ring to its minimum running radius and maximum and minimum stresses for all symmetrically located fibres are equal.

5. Additional effects and model adaptation

Theory as discussed until now points towards the following equation to obtain equal stresses at the inner and outer ring fibres:

$$R_{curl} = 2 \cdot R_{\min} \qquad (7)$$

Up to this point however a limited number of influences has been taken into account:

1. Pre-bend process (mechanical residual stress introduction).
2. Relaxation and nitriding process (additional residual stress influence).
3. Bending of rings with rectangular cross-sections based on simple beam theory.

The focus of these influences has been laid on stress levels in the longitudinal direction using a one-dimensional analysis. In practice the following phenomena will have an additional effect on the stress level in the ring. Some of them require a two- or even a three-dimensional analysis.

1. Contact between element and inner ring
2. Crowning radius of the ring in transverse direction
3. Anticlastic effect leading to cross-section variations during bending

These phenomena are discussed below.

5.1. *Contact between elements and inner ring*

When the inner ring is at the pulley, its overall running radius is locally disturbed by the actual surface the ring is in contact with. This surface made up out of elements forms a polygon surface as illustrated in **figure 8**. It influences the stress profile in the ring by the following two effects:

- Contact stress in the local contact point between ring and element.
- Additional stress from local bending.

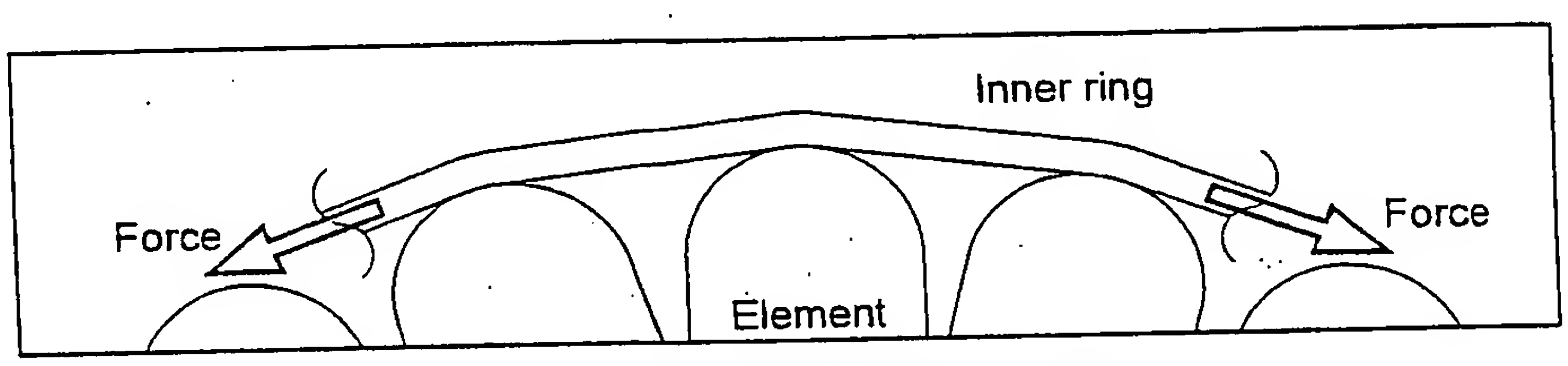


Fig. 8: Contact between element and ring: polygon effect. (Figure is not on scale).

Contact stresses occur at the inner radius of the inner ring where the ring is in contact with the elements. In the contact friction occurs resulting from the ring sliding over the elements.

The element - ring contact can also lead to additional local stresses from bending in the ring. As the ring follows the local contact geometry, it experiences a smaller bending radius than the overall running radius. This causes increased compression stress at the inner fibre and increased tension stress at the outer fibre of the ring. In theory also local straightening of the ring can occur between two elements as can be seen in figure 8.
At the straight part between the pulleys the effect is not present. Stress amplitudes between the situations straight (not on pulley) and bent (on pulley) therefore will increase for inner and outer fibre.

The possibilities for stress reduction by means of measures in the element - ring contact fall outside the scope of this paper. The subject remains an important research item.

5.2. *Crowning*

After production the ring is not flat in the transverse direction. This effect is referred to as crowning. The surface curvature is approximated with a radius R_{crown} as defined in **figure 9**.

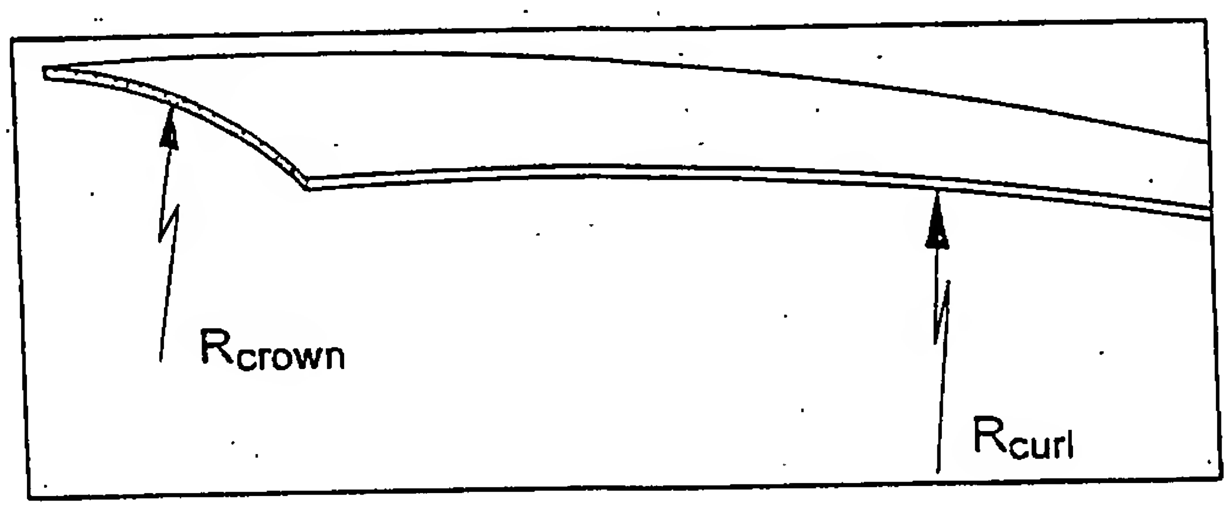
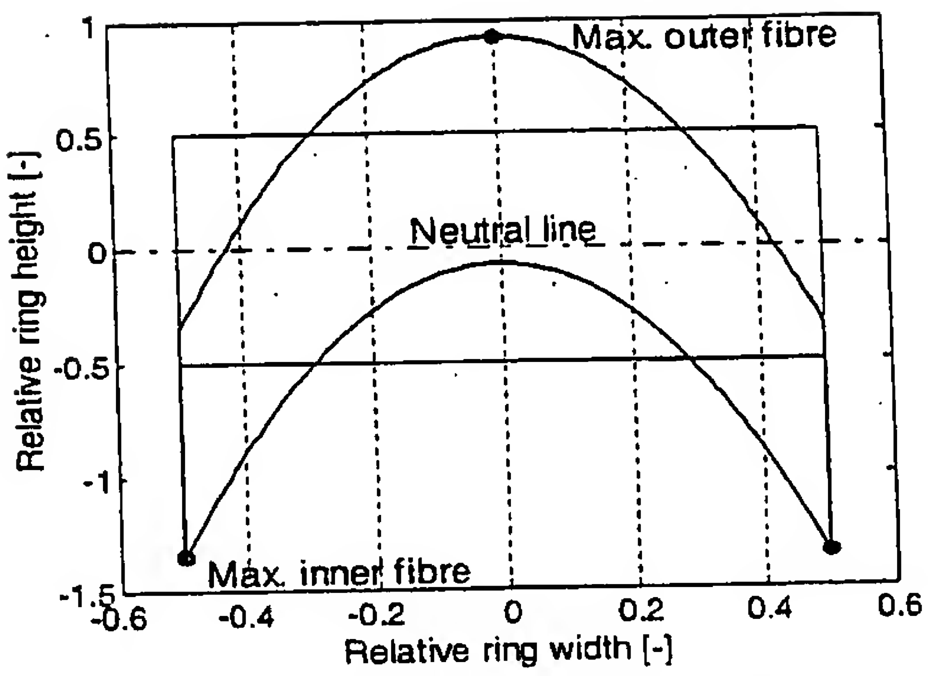


Fig. 9: Definition crowning radius.

For small crowning radii, the cross-section of the ring can no longer be considered rectangular. **Figure 10** shows increasing maximum fibre distances for crowned rings. The changing fibre distances lead to a different moment of inertia. As the cross-section no longer is symmetrical around the horizontal neutral line, the minimum moment of resistance W is different for inner and outer fibre. This can be seen in **figure 11**.
At the pulley the ring experiences a pre-determined bending radius. The bending moment depends on bending radius and moment of inertia according equation (1). Fibre distances and bending radius determine the stress levels in the ring. The increased fibre distances lead to higher stresses for crowned cross-sections in comparison to rectangular cross-sections.



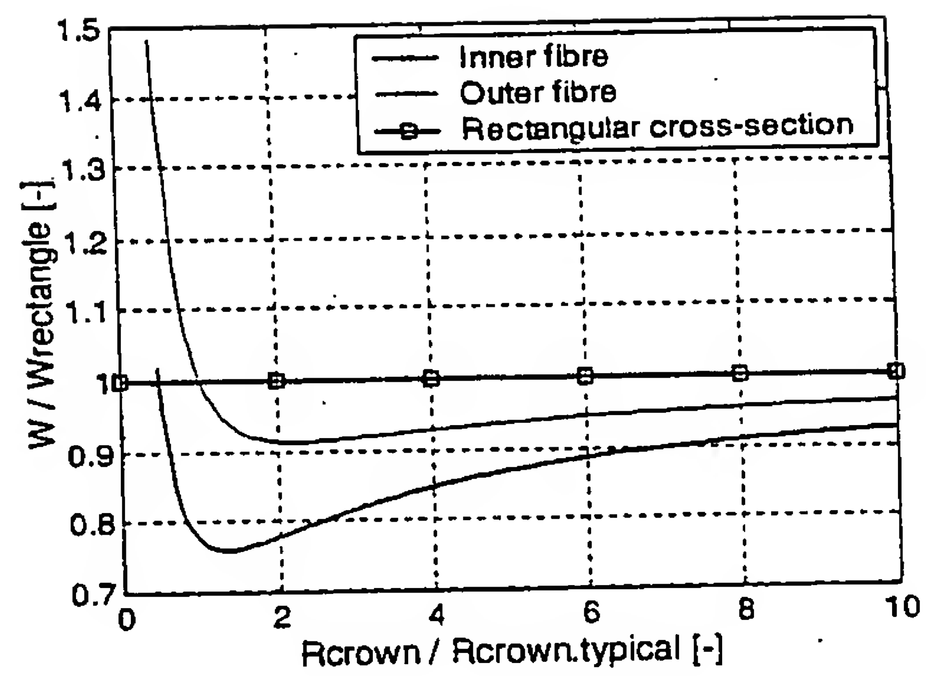


Fig. 10: Ring contour rectangular versus crowned cross-section.

Fig. 11: Relative minimum moment of resistance for inner and outer fibre as a function of crowning radius.

From **figure 12** the ultimate strains in longitudinal direction can be derived based on simple beam theory. For the moment an initially straight beam with crowned cross-section bent at a radius R is assumed.

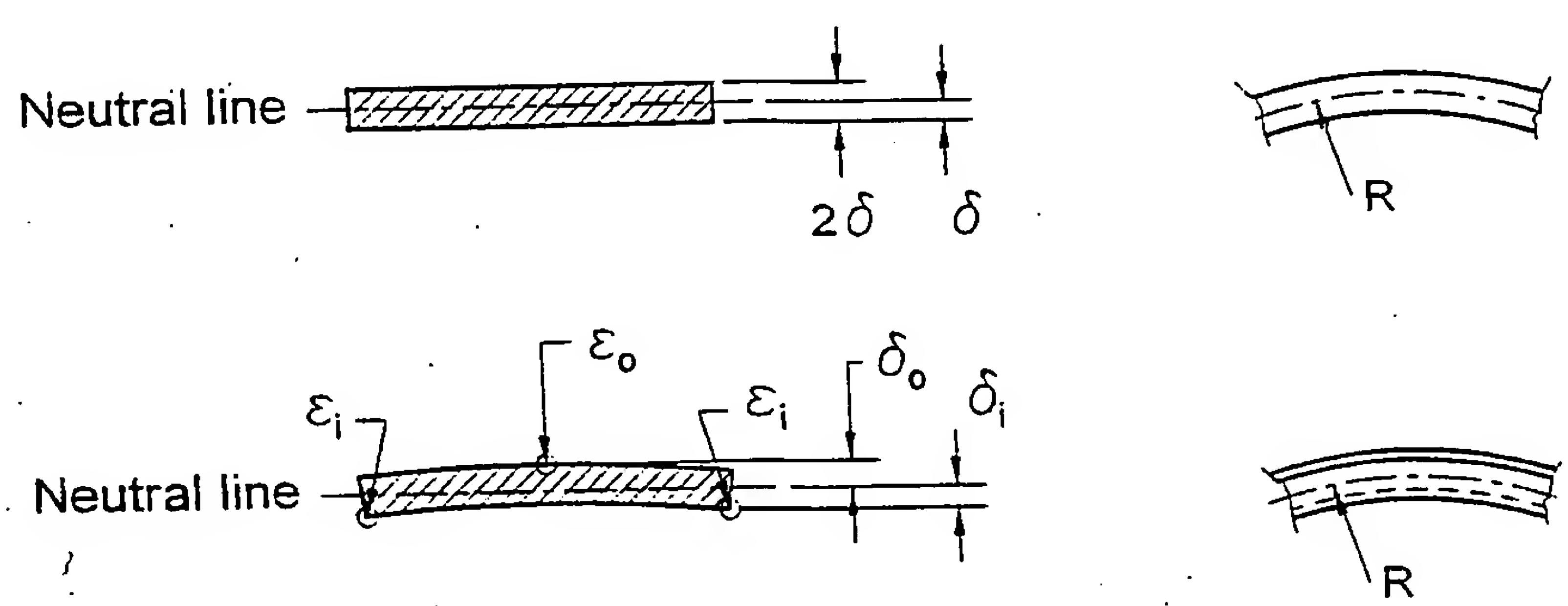


Fig. 12: Rectangular and crowned cross-section bent at radius R.

The equations for the ultimate strains in the crowned cross-section are:

$$\varepsilon_i = \frac{-\delta_i}{R} \tag{8}$$

$$\varepsilon_o = \frac{\delta_o}{R} \tag{9}$$

Where:

$\delta_{i,o}$ = maximum fibre distance at inner and outer surface [mm]

$\varepsilon_{i,o}$ = longitudinal strain at inner and outer maximum fibre distance [-]

Equation (5) can now be rewritten. Again the condition of equal maximum tension stress for inner and outer surface is used. Under the assumption that the residual stress is σ_{pb} at both locations, the equation becomes:

$$-\sigma_{pb} + \frac{\delta_i \cdot E}{R_{curl}} = -\sigma_{pb} - \frac{\delta_o \cdot E}{R_{curl}} + \frac{\delta_o \cdot E}{R_{\min}} \tag{10}$$

From this equation the pre-bend factor f_{pb} can be derived:

$$R_{curl} = \frac{\delta_i + \delta_o}{\delta_o} \cdot R_{\min} \tag{11}$$

$$f_{pb} = \frac{\delta_i + \delta_o}{\delta_o} \tag{12}$$

For a rectangular cross-section where $\delta_i = \delta_o$, the pre-bend factor f_{pb} is 2. For crowned cross-sections the factor depends on crowning radius and ring geometry. **Figure 13** shows equation (12) as a function of crowning radius for a typical ring geometry.

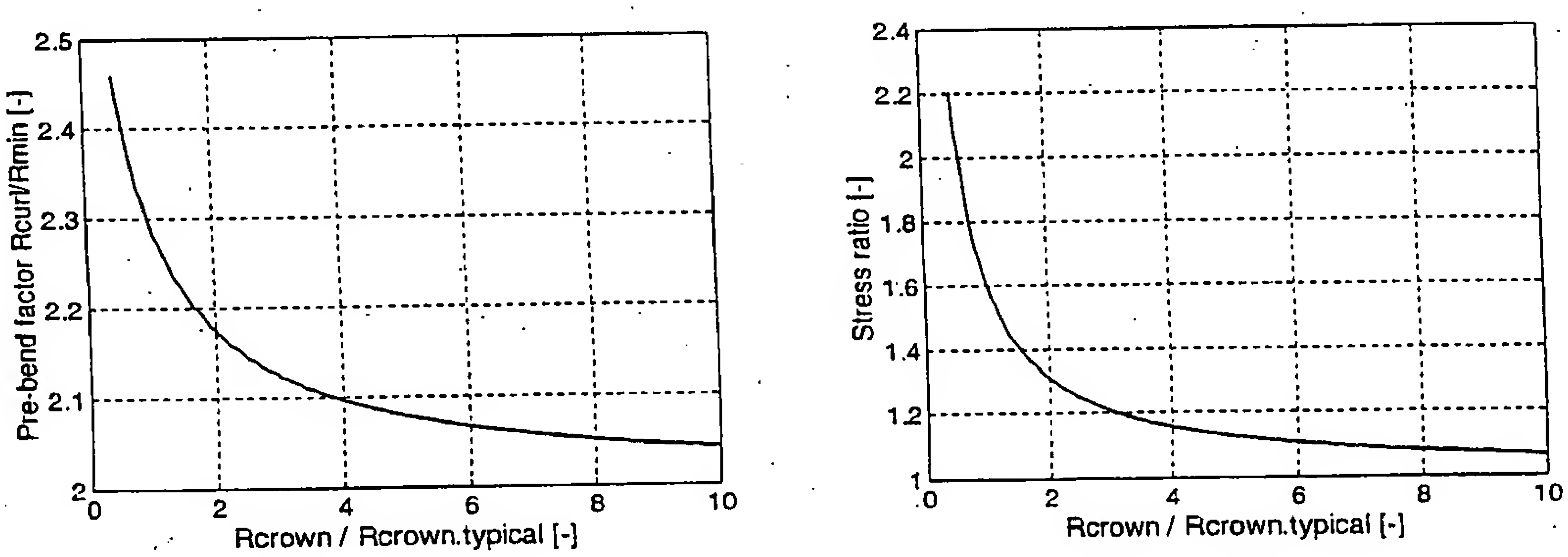


Fig. 13: Pre-bend factor f_{pb} as a function of crowning radius for a typical ring geometry.

Fig. 14: Stress ratio between a crowned and a rectangular cross-section as a function of crowning radius for the optimum pre-bend factors of figure 13. Only crowning is reviewed!

At this point stress increases caused by crowning of the ring can be estimated. **Figure 14** shows tension stress ratios between a rectangular and a crowned cross-section as a function of relative crowning radius for the typical ring geometry of figure 13.

For certain curling and crowning radius combinations with optimal pre-bend factor, stresses can be further minimised by increasing the crowning radius. In case an optimum pre-bend factor is maintained this leads to a decreasing curling radius.

It can be concluded that crowning influences the stress levels from bending in the rings and therefore directly affects the optimum curling radius. Larger crowning radii result in lower stress and smaller optimum curling radii. Crowning leads to pre-bend factors larger than 2.

5.3. *Anticlastic effect*

In case a ring with a curling radius and a crowning radius is bent, crowning changes. During bending straight the radius R_{crown} decreases and crowning becomes worse. During bending at a smaller radius than the curling radius, R_{crown} increases and crowning reduces. This behaviour, illustrated in **figure 15**, can be explained by the anticlastic effect.

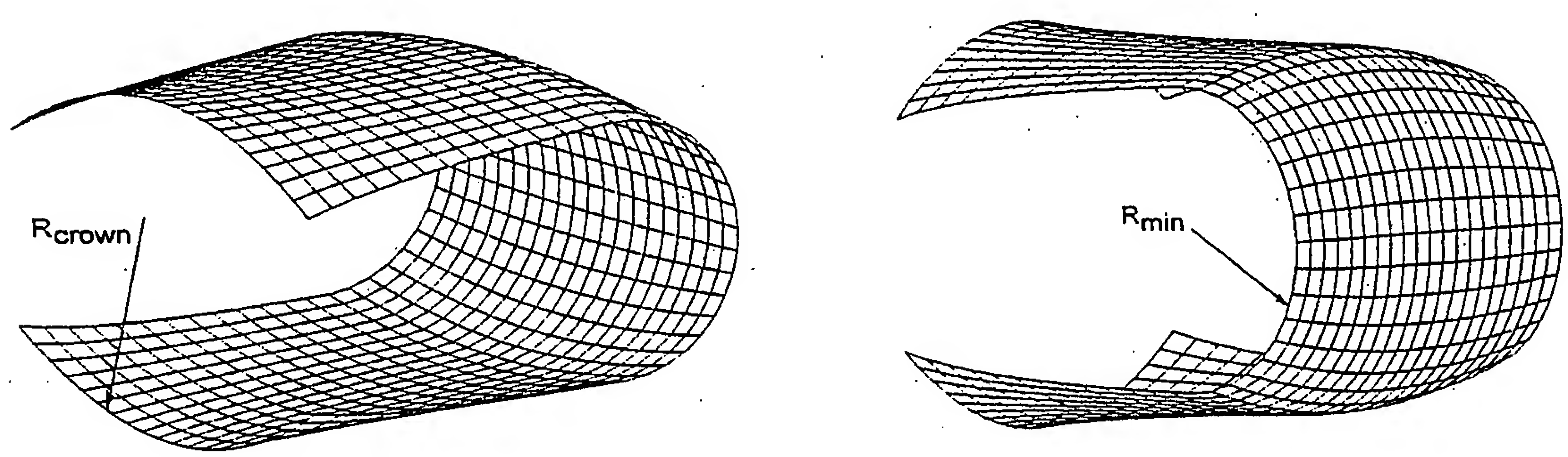
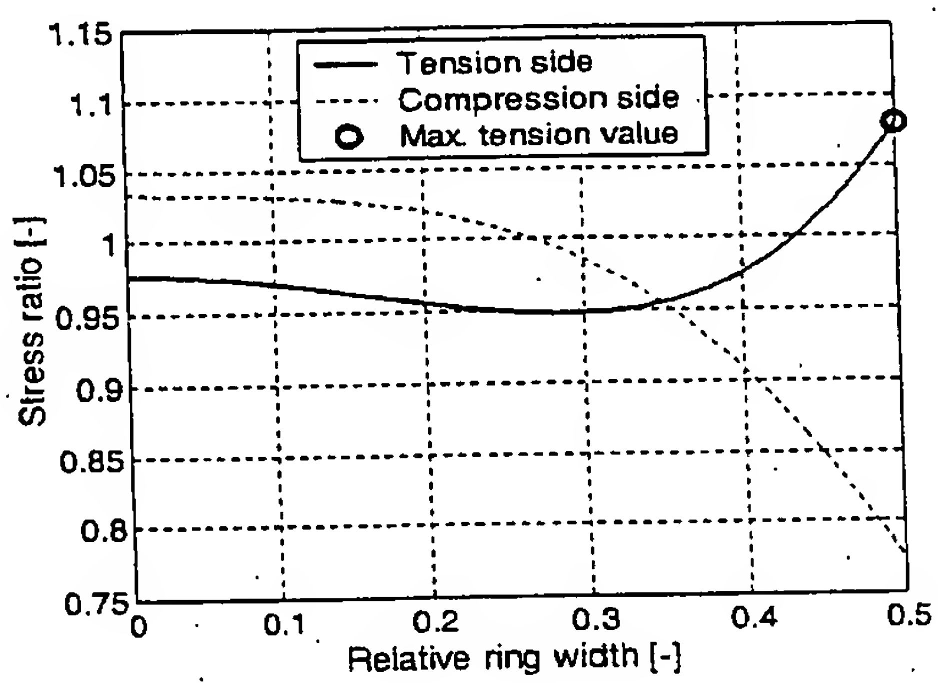


Fig. 15: Varying crowning radius during bending and straightening.

This effect, caused by transverse contraction, introduces crowning changes during bending. The effect is described in literature for straight plates with a rectangular cross-section [4]. Also described here are the differences in stresses on the surface for anticlastic theory and simple beam theory. The ratio between these two is given for a rectangular cross-section in **figure 16**. Shown are the ratios for tension and for compression stress, both in longitudinal direction.

Typically tension stress increases at the edges of the ring. Compression stress decreases at these locations.

Figure 17 shows the deflection of the neutral line for the initially rectangular cross-section.



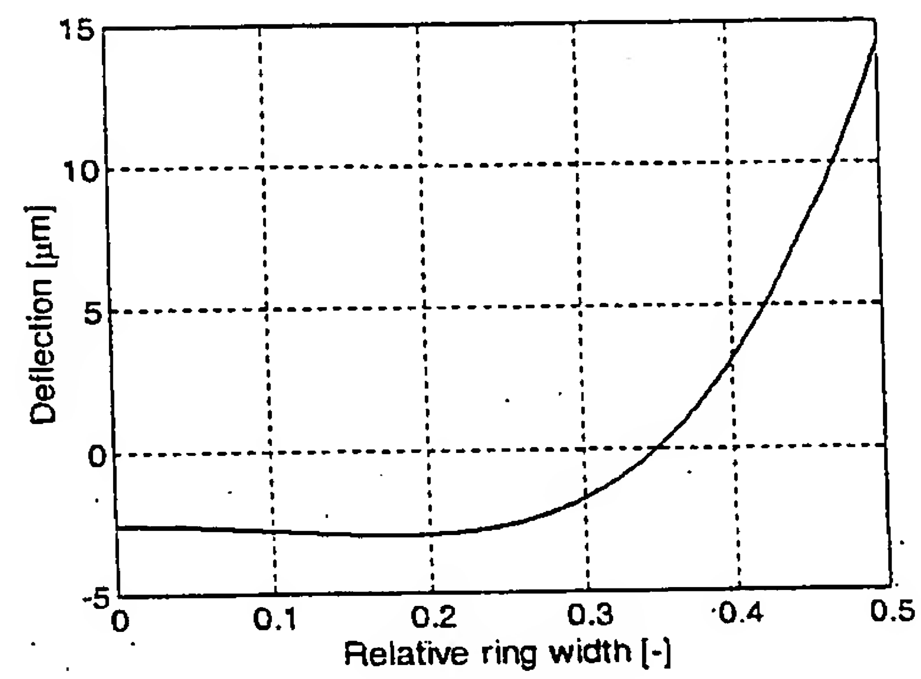


Fig. 16: Stress ratio anticlastic versus beam theory for a typical rectangular cross-section bent at R = 30 [mm].
Fig. 17: Anticlastic ring deflection for a typical rectangular cross-section bent at R = 30 [mm].

Clearly crowning radius and stress are influenced by the anticlastic effect. The effect can be introduced for crowned rings using two factors applicable for the locations of interest. This extends equation (10) as follows:

$$-\sigma_{pb} + f_i \cdot \left(\frac{\delta_i \cdot E}{R_{curl}}\right) = -\sigma_{pb} - f_o \cdot \left(\frac{\delta_o \cdot E}{R_{curl}} - \frac{\delta_o \cdot E}{R_{\min}}\right) \tag{13}$$

Where:

f_i = stress factor anticlastic effect, maximally loaded inner fibre during straightening [-].

f_o = stress factor anticlastic effect, maximally loaded outer fibre during bending [-].

With the introduction of these factors the assumption of simple beam theory is left. Changes in maximum fibre distances, caused by deforming cross-sections and varying neutral lines, are now included.

The factor f_i is applicable for the edges of a ring with crowned cross-section. Tension will be maximal here because fibre distances are maximal. The factor f_o is applicable for the middle of the ring for the same reason.

From equation (13) follows:

$$R_{curl} = \frac{f_i \cdot \delta_i + f_o \cdot \delta_o}{f_o \cdot \delta_o} \cdot R_{\min} \tag{14}$$

To investigate the factors f_i and f_o, three-dimensional Finite Element Method (FEM) calculations were performed with varying parameters R_{min}, R_{crown} and R_{curl}. Anticlastic effects were included. **Figure 18** and **figure 19** show some results for identical curling radius and two different crowning radii. Tension stresses from FEM analysis are compared with tension stresses calculated with equation (10) in which anticlastic effects are not included.

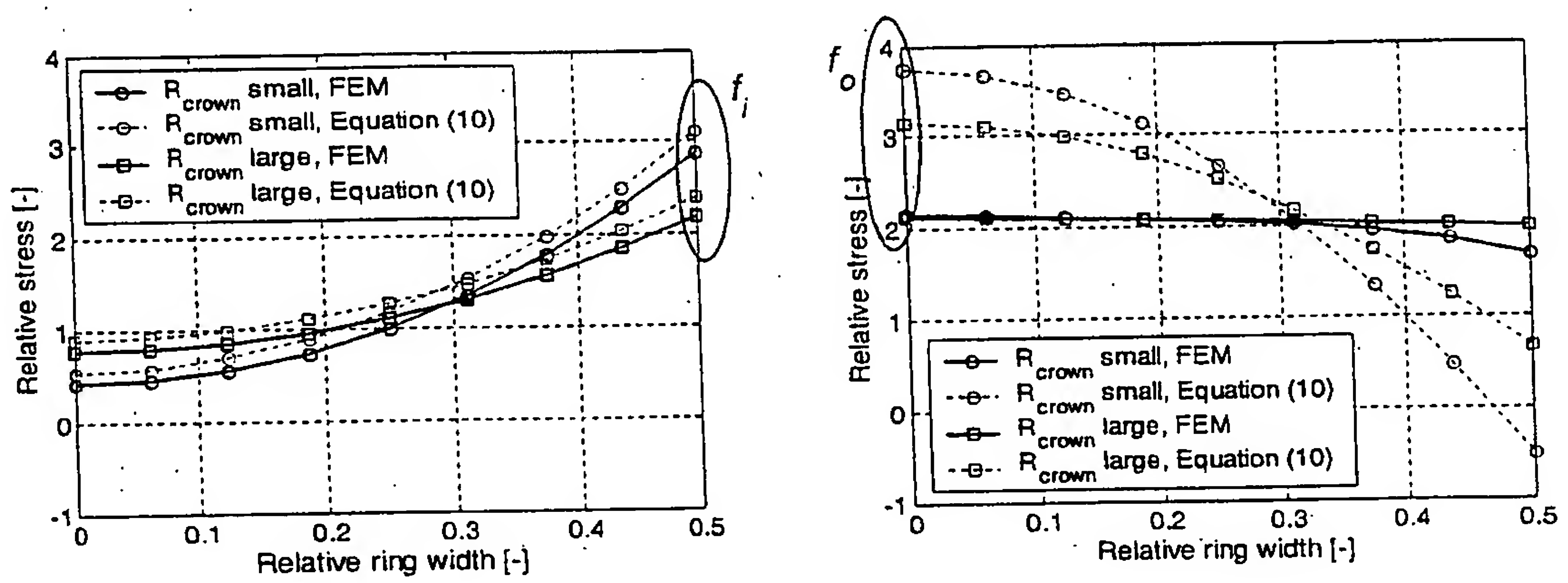


Fig. 18: Stress results longitudinal direction, inner surface during **bending straight**.
Fig. 19: Stress results longitudinal direction, outer surface during **bending to a minimum radius**.

The influence of anticlastic deflections on stresses during straightening of the ring is relatively small as can be seen in figure 18. The influence of anticlastic deflections on stresses during bending to the minimum radius is much larger. Figure 19 shows that the stress increase caused by crowning is partly undone by the anticlastic effect.
Figures 18 and 19 show that maximum tension stresses still can be found at the inner surface edges for rings bent straight and at the outer surface middle for rings bent to a smaller radius.

Comparison of results from FEM analysis and equation (10) learns that in the considered typical ranges for crowning, curling and bending radii the factor f_i is rather independent on crowning radius. Factor f_o is dependent on crowning radius. FEM analysis shows that both factors depend on curling radius in the same degree, which also can be recognised in **figure 20**. It leads to the conclusion that f_i/f_o merely is dependent on crowning radius.

The factor f_o and thus f_i/f_o is practically independent on the minimum bending radius R_{min}. Variations of R_{min} within its typical range show no significant differences. This is in accordance with literature [4]. Figure 20 contains an example of an increased minimum bending radius.

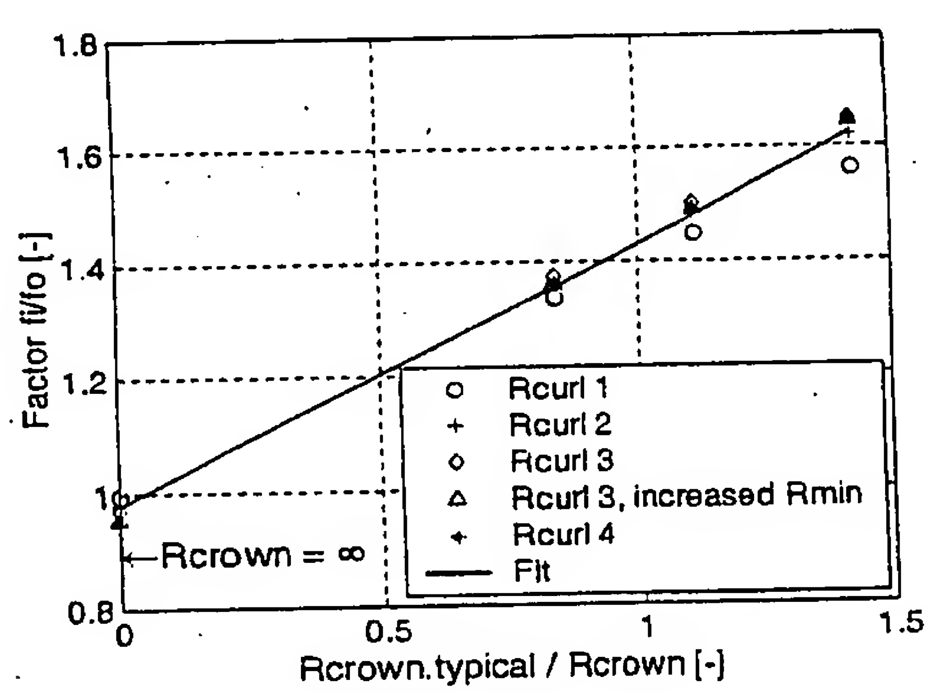
Knowing this, we can write:

$$f_{pb} = \frac{\left(\frac{f_i}{f_o}\right) \cdot \delta_i + \delta_o}{\delta_o} \tag{15}$$

For a typical ring geometry the factor f_i/f_o by approximation is linearly dependent on the reciprocal of the crowning radius. This is depicted in figure 20.

Note that the anticlastic effect can cause a shift in the location of maximum tension stress for larger crowning radii. The approach above where it is assumed that the maximum tension stress lies in the middle of the ring (figure 19) is then no longer valid. For a rectangular ring cross-section for instance, the location of maximum stress during bending does not lie in the middle. It lies at the edges as shown in figure 16.

The range of crowning radii where the maximum tension stress from bending remains located in the middle of the ring however lies within the typical range of crowning radii. Equation (15) is an acceptable approximation here.



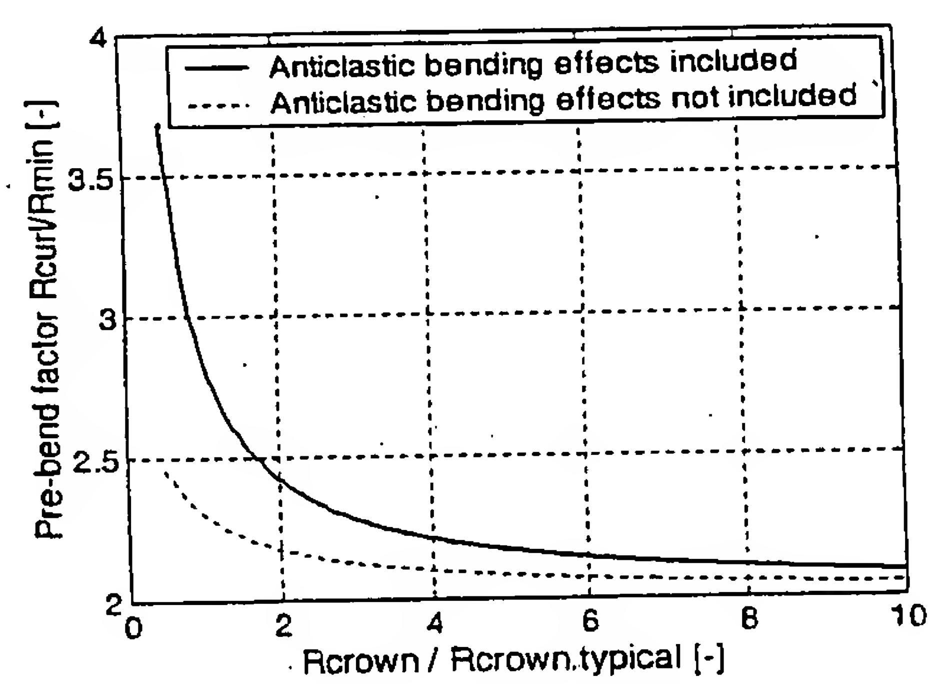


Fig. 20: FEM analysis results: factor f_i/f_o as a function of inverse crowning radius for a typical ring geometry.

Fig. 21: Approximation pre-bend factor f_{pb} as a function of crowning radius, anticlastic effects included.

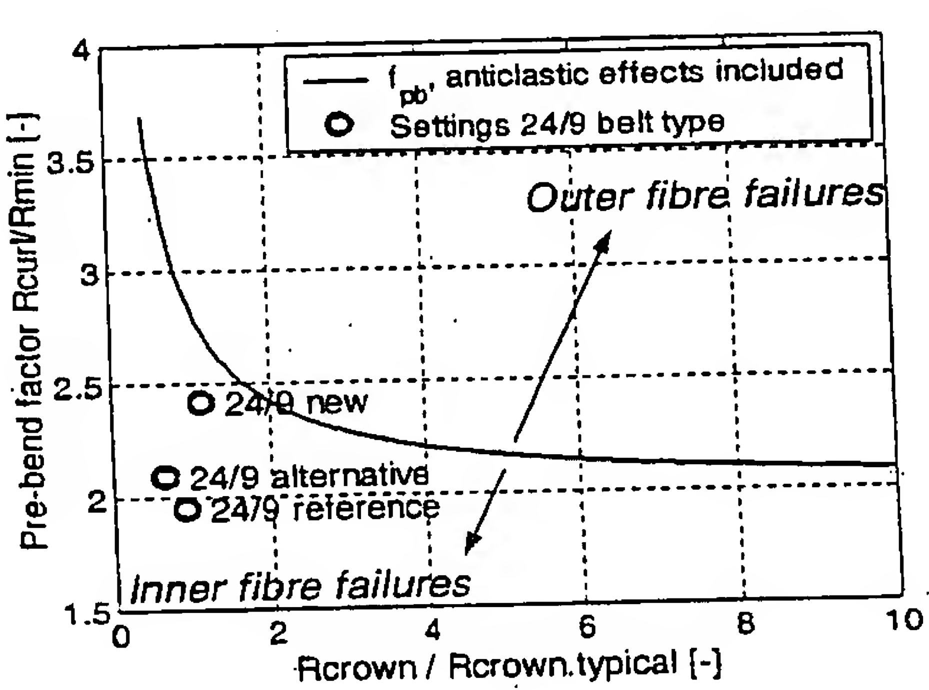
Figure 21 shows equation (15) for a typical ring geometry. Also the pre-bend factor f_{pb} without anticlastic influences from equation (12) is included.

It is concluded that the anticlastic effect influences tension stresses in a crowned ring. FEM analysis shows that the one-dimensional model presented in the previous section is allowed in case a multiplication factor is included describing the influences of the anticlastic effect. This factor mainly brings a reduction of tension stresses at the outer fibre into account, estimated too high by the one-dimensional model. The anticlastic effect results in larger optimum curling radii.

6. Test definition

Tests on rings with the current process settings show that for belt failures caused by ring fatigue, fracture initiation predominantly occurs at the inner radius of the inner ring in the belt. As shown, curling radius, crowning radius and element - ring interaction influence the inner fibre load. With the latter falling outside the scope of this investigation a test is done with changed curling and crowning radii.

To validate the described curling and crowning radius influences, tests with two belt types have run. Tested are the 24/9 belt type with 24 mm element width and 2x9 rings and the 30/12 belt type with 30 mm element width and 2×12 rings.
The settings for the rings are drawn in **figure 22** and **figure 23**. To approach the optimum settings described by theory, the new settings should result in increased curling and/or crowning radii. For production reasons it was not possible to reach an exact optimum.
Note that due to geometrical differences between the two belt types the curves defining the optimum pre-bend factor are not equal.



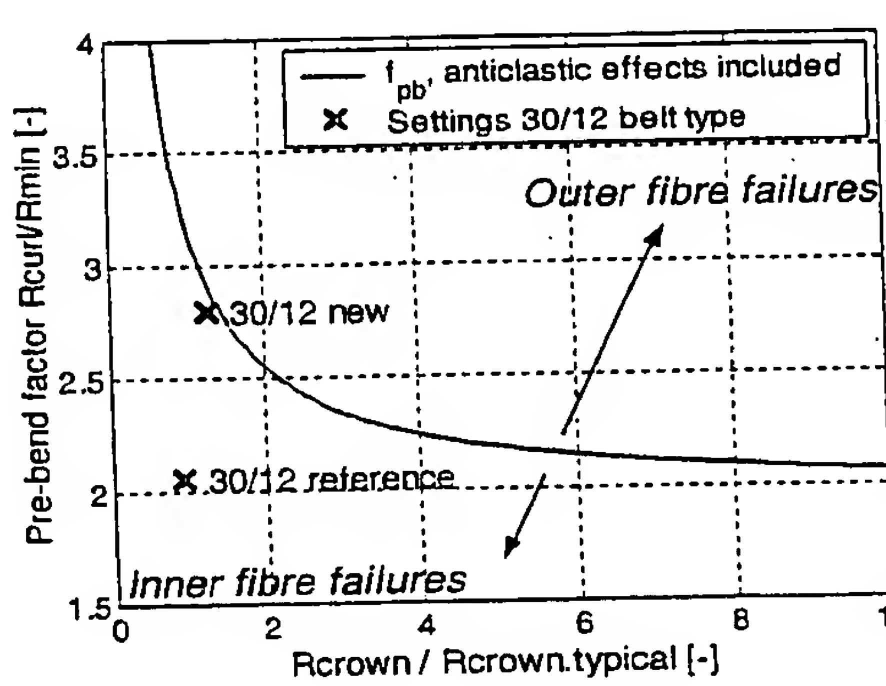


Fig. 22: Settings test data 24/9 belt application.
Fig. 23: Settings test data 30/12 belt application.

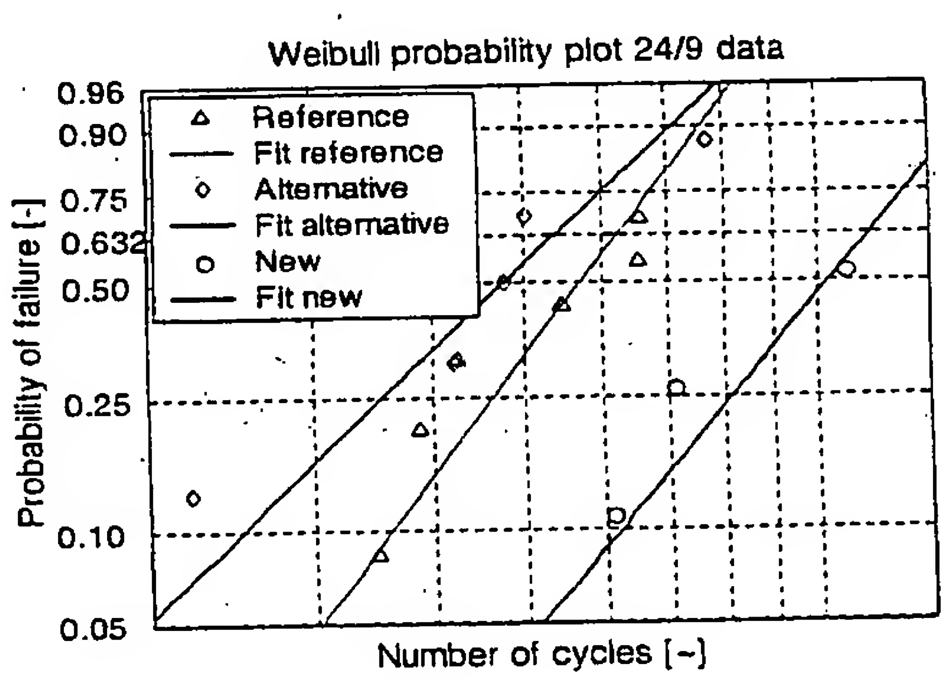
All belts were tested using an Overload Fatigue Test (OFT). This is a single ratio test on variator level during which belt and variator are tested in a belt box powered by an electric motor. It is an advanced-stress test designed to obtain life-testing data within a reasonable short period. During the test the belt is overloaded above the full load settings that occur during operation in a vehicle and which are normally applicable for Long Durability Cycle (LDC) belt box tests.

During an OFT both speed and torque are increased beyond the full load level. The variator ratio is adjusted to obtain a small running radius at the driven pulley. The clamping force is increased to maintain a normal safety. The belt is tested until ring fracture occurs which is detected using a fracture detection method.

7. Test results

The test results are shown using a Weibull plot in **figure 24** and **figure 25**. Figure 24 shows the results for the 24/9-belt application. Figure 25 shows the results for the 30/12-belt application.

For both belt types the rings with the new settings show increased life with a factor that lies around 1.7.



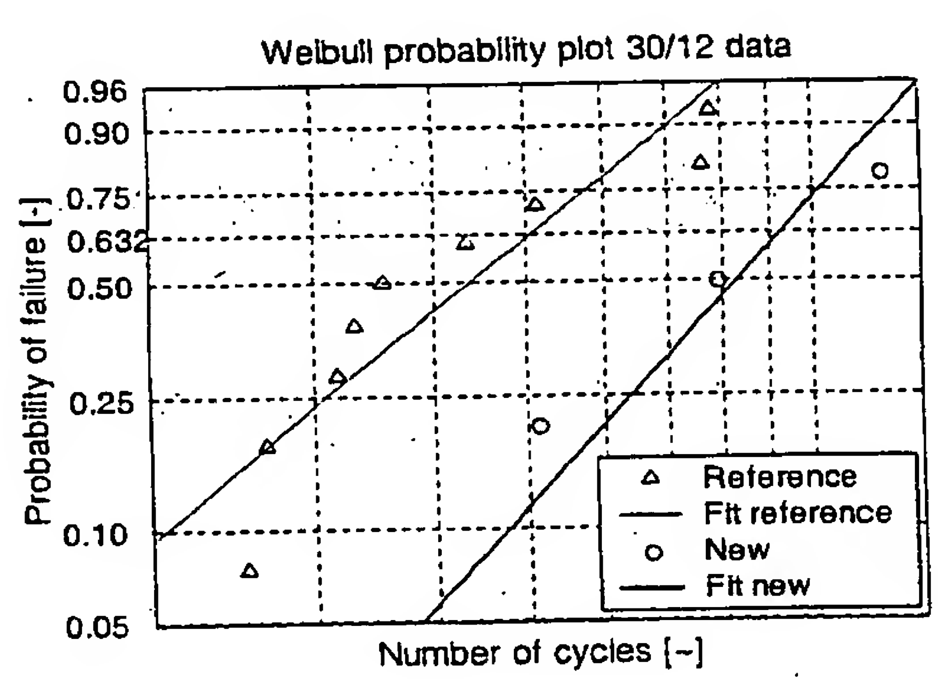


Fig. 24: Test results for 24/9 belt type data.
Fig. 25: Test results for 30/12 belt type data.

The 24/9 belts with rings produced using the alternative setting with reduced crowning radius show a slight decrease in life. All results indicate that the hypothesis of improved life for a more optimal combination of crowning and curling radius can be accepted.

For some of the new 30/12 belts crack initiation was found to lie in the middle of the outer surface of the inner ring, supporting the theory presented in this paper.

8. Implementation in CVT applications

Based on the discussed effects, power density of the push belt CVT can be further improved by using increased crowning and/or curling radii. The new process settings are used in the new belt design and lead to a crowning and curling radius increase of about 25% compared with the current belt design.

At this moment VDT is carrying out validation tests for 5 different car/transmission manufacturers from Japan, Europe and the U.S.A. to secure the introduction of the new belt design into new CVT applications. Production for these applications starts in 2002, 2003 and 2004 respectively.

The first new application equipped with the new belt design is the Nissan Murano. The Nissan Murano has been introduced to the world at the New York International Autoshow in spring 2002 and will be available in October 2002 with push belt CVT. Vehicle specifications for engine and transmission can be found in **table 1**.

Nissan Murano			
Engine		*Transmission*	
Type	3.5 ltr V6	Torque convertor	Yes, T_c-factor of 2.0
Max. engine power	180+ kW / 240+ Hp at 6000 rpm	Transmission type	Push belt CVT
Max. engine torque	350 Nm at 4000 rpm	Belt type	30/12 new design
		Ratio coverage	5.4
		Max. belt torque	500+ Nm

Table 1: Drive line specification Nissan Murano with push belt CVT.

With an engine specification of 350 Nm/180+ kW and a torque convertor with a torque ratio T_c of 2.0, the Nissan Murano CVT presently is the belt CVT with the highest power and torque capacity.

9. Conclusions

1. Test results support the improved life hypothesis of push belt rings through optimised curling and crowning radii of the rings. The results show an improvement in life testing times of about 70%. More in general this can be interpreted as a considerable improvement in power density for the 24/9 and 30/12 belt designs.
2. Given the good results, the new settings have been implemented in a new belt design leading to a further power density increase of the push belt CVT. The first transmission with the new belt design is the new 350 Nm/180 kW Nissan Murano application where a new 30/12 push belt is used.
3. Pre-bending of push belt rings during production introduces residual stress in the ring material that helps to decrease ring stress caused by bending during variator operation. This allows a higher power density for the push belt.
4. For rings with rectangular cross-section a pre-bend factor $R_{curl}/R_{min} = 2$ theoretically leads to minimal tension stresses in the rings in case anticlastic effects are neglected.
5. In this paper a one-dimensional ring model is described for optimisation of the curling radius of the rings. The model takes the effects of crowning and anticlastic curvature into account.
6. Simulations using the new model on crowned ring cross-sections influenced by anticlastic curvature effects reveal that pre-bend factors increase and lead to values significantly larger than 2. Effectively leading to larger optimum curling and /or crowning radii.

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